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SMALL HIGH-PRESSURE LIQUID OXYGEN TURBOPUMP

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LIQUID OXYGEN TURBOPUMP Final Report
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A. Csomor

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FOREWORD

The work herein was conducted by the Advanced and Propulsion Engineering and the Engineering Test personnel of Rocketdyne, a division of Rockwell International, under Contract NAS3-17800 from August 1973 to July 1978. Mr. R. Connelly, Lewis Research Center, was NASA Project Manager. At Rocketdyne, Mr. H. Diem as Program Manager, and Mr. A. Csomor as Project Engineer were responsible for the technical direction of the program.

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Mark 48-0 LOX Turbopump

SUMMARY

The objective of this program was to establish the technology for small, high-pressure liquid oxygen (LOX) pumping capability. Turbopumps in this category are needed for applications in small, high-performance, reusable, versatile, staged-combustion rocket engines. To accomplish this objective, analysis and design efforts were expended to produce specifications and shop drawings in sufficient detail to permit fabrication of test hardware.

To obtain high performance and minimize weight, the rotor speed was established at 7330 rad/s (70,000 rpm). The pump design included a single-stage centrifugal impeller preceded by an axial-flow inducer to reduce the net positive suction head (HPSH) requirements. Rotor axial thrust control was provided by incorporating a self-compensating, double-acting balance piston as an integral part of the impeller rear shroud. Power for the pump was developed by a single-stage, partial-admission turbine using the combustion products of liquid hydrogen (LH₂) and LOX as the propellant. The rotor was supported on two ball bearings at each end. The pump end bearings were cooled by recirculating LOX. The turbine end bearings, located outboard of the turbine disk to provide auxiliary power takeoff capability, were cooled by LH₂. Controlled gap seals were used to accomplish sealing along the rotor.

Hardware was fabricated for two complete turbopump assemblies. To provide a hot-gas source for the turbine, a gas generator was designed, fabricated, and tested.

The turbine was calibrated at Wyle Laboratories with gaseous nitrogen as the driving fluid, and a torquemeter was used to measure output. The turbine efficiency was measured at 51% at the design point.

The initial testing of the complete turbopump was performed in July-August 1976 at Lima Stand of Rocketdyne's Propulsion Research Area (PRA). Eighteen tests (257 seconds) were conducted on one turbopump assembly, with LOX as the pump fluid on all but three tests. (Liquid nitrogen [LN₂] was initially used to verify integrity.) The turbine was propelled by ambient-temperature gaseous hydrogen on seven tests, and by hot gas on the remaining tests. Speeds in excess of the design level, up to 7765 rad/s (74,191 rpm) were explored. Pump discharge pressures ranging up to 3175 N/cm² (4604 psia) were generated with flowrates up to 0.013 m³/s (193 gpm). The turbine was exposed to a maximum inlet temperature of 1133 C (2040 R).

The hydrodynamic data revealed that the pump suction performance did not meet design goals in the high flow region. Analysis of the data and design approach revealed that a larger impeller inlet area was required to improve performance. No structural problems were indicated either by the data or post test hardware condition. High temperatures noted in the pump end bearings indicated a need for higher flowrates through the balance piston and the bearings.

To improve performance, the impeller inlet area was increased and the balance piston and bearing coolant flow was routed overboard where it could be measured and controlled. In the second test series, during July 1977, five tests were conducted with a total time of 158 seconds. The testing encompassed noncavitation head-flow characterization of the pump. While critical net positive suction head (NPSH) was not defined, the pump operated without cavitation up to a suction specific speed (N_{SS}) of 24,400 which represented a major improvement over the original performance. Testing was curtailed by a mechanical malfunction which resulted in a fire in the LOX region and caused major damage to the pump hardware. The origin of the malfunction was conclusively established as the primary seal nut backing out of its installed position and blocking the overboard passage for the balance piston and bearing coolant fluid.

Design changes were made to a second set of hardware to improve the primary seal nut locking feature, and reduce the tendency of the fluid vortex to loosen the nut. Changes were also incorporated to protect the pump end bearing against high axial load from coolant pressure drop. Four additional tests (236 seconds) were conducted in May 1978 to validate the modifications and obtain hydrodynamic data. The test series was terminated prematurely by a high rotor torque condition. Disassembly revealed the high torque originating from turbine tip seal rubbing. The newly incorporated features functioned properly and the pump head-flow data corroborated the satisfactory performance obtained on the second test series.

INTRODUCTION

System studies have been conducted to determine the feasibility of developing a reusable vehicle for performing future Air Force and NASA space maneuvering missions. These studies have shown that, over the thrust range of interest, high-pressure, staged-combustion-cycle engines offer the highest specific impulse and payload capability. A review of the vehicle and engine system study results indicates that a single-bell-nozzle, staged-combustion-cycle engine at 88,964 N (20,000 pounds) thrust level is near optimum for the DOD and NASA mission requirements.

This program was initiated to provide the required LOX turbopump technology base for subsequent development of a high-performance, staged-combustion rocket engine.

Technology items of particular interest during the course of this program included establishing the fluid dynamic parameters and design details for a small-capacity, high-pressure LOX pump, and low-pressure-ratio, partial-admission turbine; operation of a balance piston with no axial rubbing features; balance and operation of a high-speed rotor; high DN bearings in LOX; hydrogen-environment embrittlement protection; and fabrication of small components with limited accessibility for generating internal passages. To provide a hot-gas source for the turbine, work was also performed on high-pressure, concentric-element, O_2/H_2 injector gas generators.

The objectives of this program were to design, fabricate, and test a high-pressure LOX turbopump capable of meeting the performance requirements of the 88,964 N (20,000 pounds) thrust, staged-combustion-cycle engine, demonstrate its basic capability, and identify any areas where additional effort due to technology limitations is required to place a future engine program on a solid basis.

Rocketdyne has assigned the designation "Mark 48-0 Turbopump" to the small, high-pressure, liquid oxygen turbopump design generated under this contract. The two terms will be used interchangeably throughout this report.

The effort performed during 1973 to 1976 which encompassed analysis and design of the turbopump, fabrication of experimental hardware, and the initial test series performed is described in an interim report published in July 1977 (ref. 1).

DISCUSSION

TURBOPUMP DESCRIPTION AND BACKGROUND

A comprehensive discussion of the MK 48-0 turbopump design requirements, analysis results, and mechanical configuration are presented in Ref. 1. For convenience, a brief summary of the significant characteristics of the turbopump is included in the following.

Turbopump Requirements

The performance requirements for the Mark 48-0 turbopump are listed in Table 1. The pump is required to deliver 16.4 kg/s (36.21 lb/sec) of liquid oxygen starting with an inlet pressure of 68.9 N/cm² (100 psia) provided by the low-pressure pump, to a discharge pressure of 2977 N/cm² (4318 psia). The propellant gas for the turbine is a mixture of free hydrogen and steam resulting from the combustion of liquid hydrogen and liquid oxygen. The gas is provided at a temperature of 1041 K (1874 R) and an inlet pressure of 2320 N/cm² (3366 psia). The total gas flowrate available is 1.34 kg/s (2.92 lb/sec). The horsepower requirement of the pump is matched by adjusting the pressure ratio across the turbine. Since turbine pressure ratio has a strong influence on the attainable engine combustion pressure in a staged combustion cycle, it is to be maintained at the lowest possible level. As noted in Table 1, the mechanical operating requirements included multiple starts with long-operating durations and potentially long-coast times between operations.

In the area of the pump, the combination of low flowrate and high discharge pressure imposed a difficult impeller fabrication task because of the relatively narrow passages required compared with the outer diameter. The desire for high efficiency, compact packaging, and light weight placed the rotor speed into the 6282 to 9423 rad/s (60,000- to 90,000-rpm) range, pushing bearing DN value to the 1.5×10^6 mm rpm limit noted in the Design Ground Rules (Appendix A). The bearing operation at high DN values in a turbopump installation, as well as the dynamic behavior of the rotor at high speeds, needed to be demonstrated. Because of the high operating speed involved, the bearings would not be able to take an appreciable axial thrust load. This condition dictated that an axial thrust balance device be employed which, in liquid oxygen, would have to be of the nonrubbing type. The operating characteristics of such a device also required evaluation.

In the turbine, the low-pressure ratio (approximately 1.4) and low arc of admission (28%) presented a combination for which no empirical data were available. Performance predictions based on calculations needed to be validated or modified by measured performance data.

From a structural consideration, the requirement for 300 thermal cycles was significant in that it established low-cycle-fatigue criteria and eventually necessitated incorporating a liner in the turbine manifold to limit the maximum thermal gradients in structural walls.

TABLE 1. LIQUID OXYGEN TURBOPUMP NOMINAL DESIGN CONDITION

	Metric Units	English Units
Turbopump		
Capable of operation at pumped-idle conditions (5 to 10 of full thrust)		
Off-design operation	±20% Q/N at full thrust down to 30% Q/N at 20% N	
Number of start-stop cycles	300	
Time between overhaul	10 hours	
Pump		
Type	Centrifugal	
Propellant	Liquid oxygen	
Inlet pressure	68.9 N/cm ²	100 psia
Inlet temperature	90-95.5K	162 to 172 R
Discharge pressure	2977 N/cm ²	4318 psia
Mass flow	16.4 kg/s	36.21 lb/sec
Number of stages	One	
Turbine		
Working fluid	H ₂ -O ₂ combustion products (H ₂ x H ₂ O)	
Inlet temperature	1041	1874 R
Inlet pressure	3220 N/cm ²	3366 psia
Pressure ratio	Minimum necessary to develop pump horsepower requirements.	
Flowrate	1.34 kg/s	2.92 lb/sec
Number of stages	One	
Type	Partial admission	
Service life between overhauls:	*300 Thermal cycles or 10 hours accumulated run time	
Service-free life	*60 Thermal cycles or 2 hours accumulated run time	
Maximum Single Run Duration:	2000 s	
Maximum time between firings during mission:	14 days	
Maximum time between firings during mission:	1 minute	
Maximum storage time in orbit (dry):	52 weeks	
Thermal cycle defined as engine start (to any thrust level) and shutdown		

In addition to the performance criteria noted in Table 1, the contract work statement included certain ground rules relating primarily to the structural analysis and mechanical design of the turbopump. These ground rules are enclosed in Appendix A.

Turbopump Description

The mechanical configuration of the small, high-pressure, liquid oxygen turbopump is illustrated in Fig. 1, with significant parts identified. The top assembly requirements are established on Rocketdyne drawing number RS009820E, which is included in Appendix B. The design was given the Rocketdyne internal designation of Mark 48-0.

Liquid oxygen is introduced to the pump through the axial-flow inlet of 4.214 cm (1.659 inch) diameter and passes through a four-bladed, constant-outer-diameter, tapered-hub inducer which raises the pressure to an intermediate level. From the inducer the liquid proceeds into a centrifugal impeller containing four partial and four full blades. Subsequently, it is diffused in a radial diffuser which incorporates 13 guide vanes. Downstream of the diffuser, liquid oxygen is collected, further diffused in a volute section, and delivered through a single 2.54 cm (1.00 inch) diameter duct.

Hot gas to the turbine is admitted through a scroll-shaped, constant-velocity inlet, lined with a 1.57 mm (0.062 inch) metal liner to maintain the thermal gradients across the structural walls at an acceptable level. The inlet duct diameter is 3.1 cm (1.22 inches). The active arc of the partial-admission nozzle extends over 1.8 rad (103 degrees) or 28.6% of the circumference, and it includes seven flow passages. The gas is fully expanded through the nozzle after which it passes through a single row of unshrouded impulse-type blades (79 blades) of the rotor. The exhaust gas is directed through a row of stationary vanes which guide the gas toward a single radial exit duct of 3.81 cm (1.50 inches) diameter.

The pump shaft and the turbine disk are designed as an integral part. On the outboard end, a stub shaft is used with a stud and nut to extend the rotor. Two pairs of angular-contact, 20-mm ball bearings are used to support the rotor. The pump-end bearings are cooled by recirculating liquid oxygen through them. The outboard shaft seal is pressurized with liquid hydrogen, and the leakage toward the outboard side is used as bearing coolant. A small amount of liquid hydrogen is bypassed around the seal and introduced to the bearing directly as a redundant source of coolant. The bearings in each pair are axially preloaded against each other with Belleville springs to prevent ball skidding. The turbine-end bearings are free of other axial loads. The outer-race sleeve of the pump-end bearings is axially retained so that the bearings absorb rotor axial thrust during transient periods when the balance piston does not control the rotor axial position.

Under conditions other than early transient stage during startup or at the end of shutdown, the rotor axial thrust is neutralized by a self-compensating balance piston. The rotating member of the piston is the rear shroud of the impeller. To operate the piston, high-pressure liquid oxygen from the

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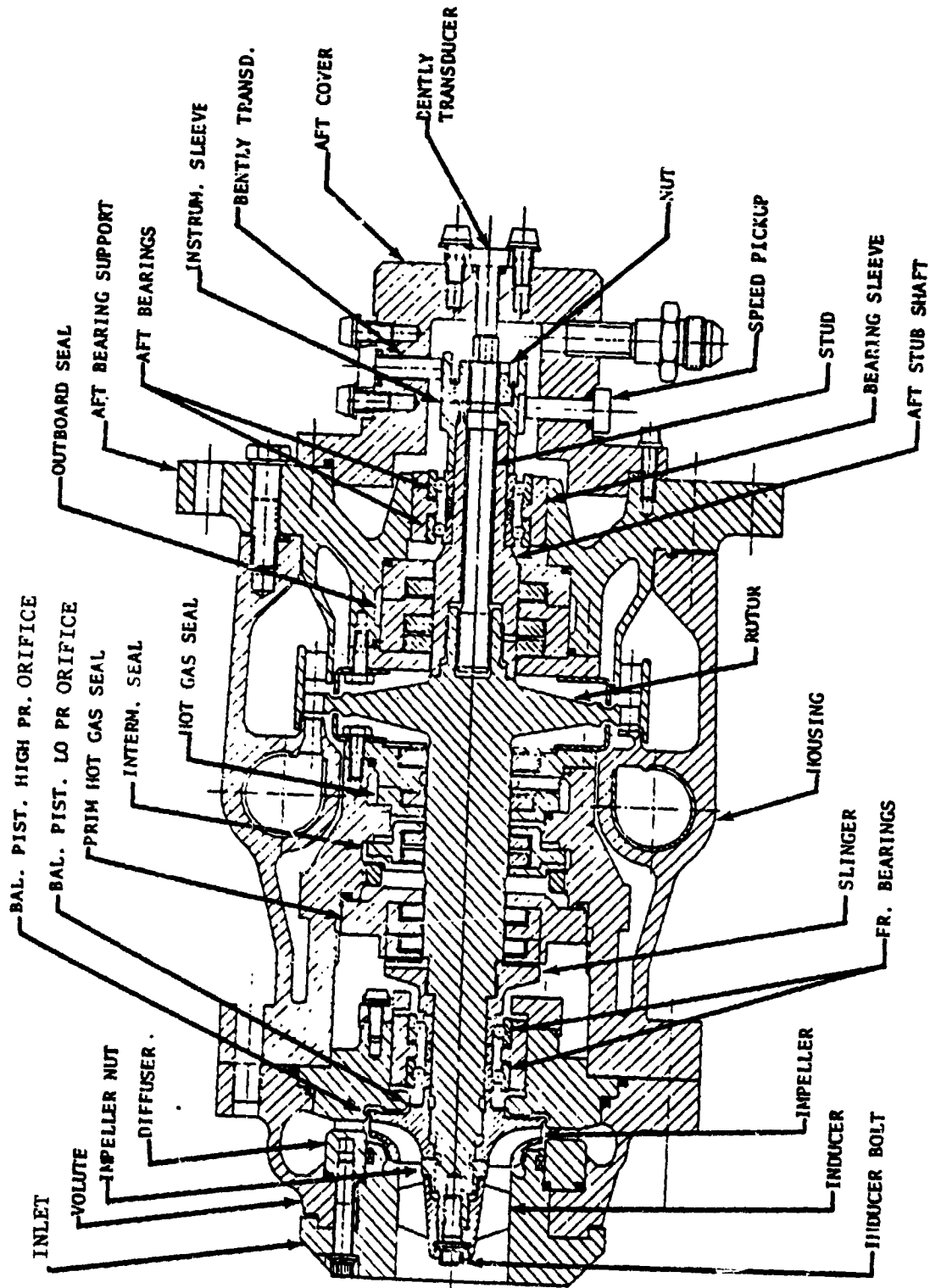


Figure 1. Mark 40-0 Turbopump

impeller discharge passes through a high-pressure orifice located at the outer diameter of the impeller into the balance cavity. From the cavity, the liquid passes through a low-pressure orifice near the impeller hub into the sump. From there the liquid oxygen is returned to the eye of the impeller through axial passages in the diffuser vanes and radial holes in the diffuser and inlet. Thrust-compensating effect is achieved by virtue of the fact that the high- and low-pressure orifice openings vary with the axial position of the rotor, and the pressure force on the rear shroud of the impeller varies correspondingly; e.g., an unbalanced load toward the pump inlet causes a reduction in the high-pressure orifice gap and an increase in the low-pressure orifice gap. This, in turn, causes a reduction in the pressure force of the impeller rear shroud, introducing a compensating load change.

Because of the danger of explosion when rubbing in liquid oxygen, the balance piston orifices were designed as noncontacting type, formed by the axial proximity of close clearance, 0.038-mm (0.0015-inch) average, diametral, cylindrical surfaces.

To preclude mixing liquid oxygen from the pump with the combustion products from the turbine, the two regions are separated by three dynamic seals. All three seals are of the controlled-gap type, with two seal rings in each. The controlled-gap concept was selected for this application primarily because it has low-drag torque, a "must" for idle-mode starts. This concept also minimizes power absorption during steady-state operation, and permits very long service life. Pump fluid is contained by the primary LOX seal. The oxygen which flows past this seal is drained overboard from the cavity formed by the primary and intermediate seals. A slinger containing pumping ribs was included upstream of the primary LOX seal to reduce the pressure at the seal gap to a level that will vaporize the fluid. The objective was to reduce the mass flowrate through the seal with this technique.

On the turbine side, because of the high pressure involved, sealing and drainage was accomplished in two steps. An overboard drain was included downstream of the first ring, which reduces the pressure between the two rings to 79 N/cm^2 (115 psia). The small amount of turbine gas which leaks past the second ring is drained overboard with a drain cavity pressure of approximately 14 N/cm^2 (20 psia).

To provide separation of the pump and turbine fluids, an intermediate seal was incorporated between the two drain areas with a GHe purge which maintains the cavity between the two rings at 35 N/cm^2 (50 psia).

Test History

Turbine Calibration. Calibration of the Mark 48-0 turbine, to establish its aerothermodynamic performance, was accomplished with ambient-temperature GN_2 as the propellant. The rotor speeds were maintained in the range of 523 to 1885 rad/s (5000 to 18,000 rpm) to simulate the operational wheel tip speed/gas spouting velocity ratios (U/C_0).

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The testing was performed at Wyle Laboratory, El Segundo, California, during February 4 through 9 1976. A total of 11 tests were made, with GN₂ working fluid, at velocity ratio (U/C_0 , total to static) ranging from 0.115 to 0.606, and turbine speeds from 523 to 1885 rad/s (5000 to 18,000 rpm). A plot of turbine test efficiency is shown in Fig. 2. The efficiency was calculated with Lebow torquemeter torque and isentropic available energy (total-to-static) across the turbine. At a design velocity ratio of 0.343, the turbine total-to-static measured efficiency was 51% compared with a predicted value of 59.8%. Calculations show that with the measured performance the pressure ratio of the turbine would have to be increased from the design value of 1.424 to 1.54 to generate the required power level.

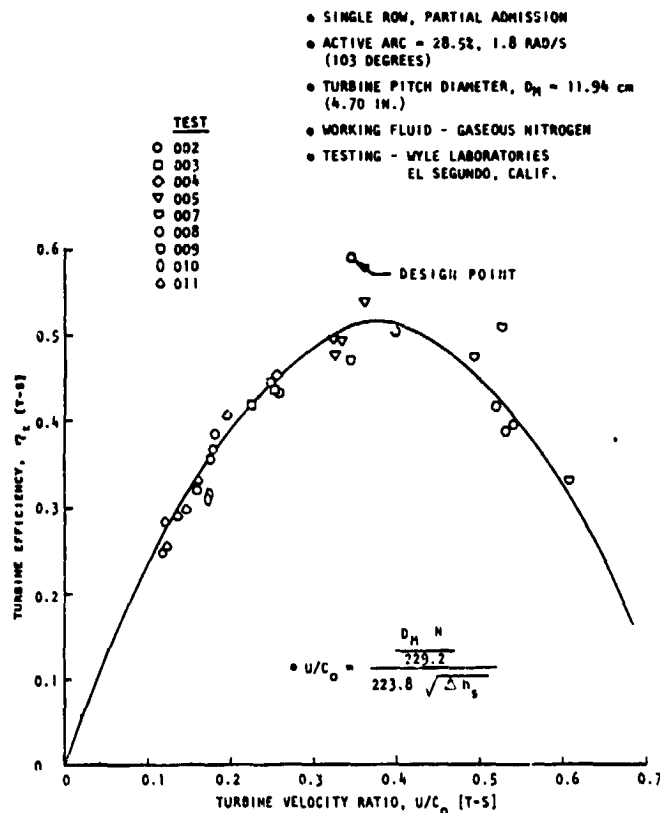


Figure 2. Mark 48-0 Turbine Performance

The combination of low-pressure ratio (1.42) and low arc of admission (28.5% of circumference) places this turbine in an operating region in which turbine technology has not been developed. Potential improvement in the performance may be realized by increasing the number of active nozzle passages and reducing the throat width to obtain the required total throat area. Depending on the engine installation, improvements in the exhaust manifolding may be possible to minimize the pressure losses charged to the turbine.

Turbopump Testing. The initial testing of Mark 48-0 turbopump P/N RS009820E, S/N 01-0, began in the Lima test stand of the Rocketdyne Propulsion Research Area (PRA) on 9 July 1976 and was concluded on 11 August 1976. A total of 18 turbopump tests for an accumulated duration of 266.8 seconds were accomplished on the turbopump assembly. The test effort was divided into two main categories: Performance mapping, using GH_2 as turbine drive media, with LN_2 and LOX as the pumped fluid; and integrity testing, using a LOX/ LH_2 gas generator as the turbine drive gas media, with LOX as the pumped fluid. Gas generator injector P/N RS005024-131, S/N 2, a coaxial five-element design, was used during the hot-fire testing. A brief description of the test performed during the initial series is presented in Table 2.

Mechanical Performance. Testing of the LOX turbopump encompassed 18 starts, with a total accumulated time of 267 seconds. The three initial tests were conducted with LN_2 as the pump fluid; in subsequent tests, LOX was used. The first seven tests were performed using ambient-temperature GH_2 to drive the turbine; in the remainder of the tests, the combustion product of LH_2 and LOX at approximately design temperature was the turbine propellant. The longest test durations conducted were 70 seconds with ambient H_2 drive and 41 seconds with hot-gas drive. The operation covered a rotor speed range of 0 to 7768 rad/s (74,191 rpm); a maximum pump discharge pressure of 3175 N/cm^2 (4606 psia); and a maximum turbine inlet temperature of 1133 K (2040 R).

Several tests were terminated by the vibration sensor device monitoring the output of the accelerometers attached to the turbopump housing. This was caused by a combination of several factors. Normally on a new turbopump several tests are required to establish its vibration signature and thus set the cutoff point at the appropriate levels. It appears that with the Mark 48-0 turbopump, this level is in the 20 to 25 g rms range in conjunction with a 2K Hz low-pass filter. Some of the early runs were terminated because the cutoff redline was set too low. In addition, the manual GH_2 feed control system employed on the first seven runs frequently resulted in slow transition through critical speed zones, with attendant buildup in vibration levels.

Bently data and accelerometer data obtained from high-frequency tapes showed increased synchronous activity at 4115, 5026, and 5528 rad/s (39,300, 48,000, and 52,800 rpm). These compared favorably with the analytically predicted critical speeds of 4723 and 5482 rad/s (45,108 and 52,363 rpm), respectively. No evidence of subsynchronous vibration was present in the data.

The measured seal drain pressures, temperatures, and flowrates were, in general, in good agreement with predicted values, indicating proper functioning of the shaft seals. During chilldown of the pump on the LN_2 tests, it was noted that the secondary hot-gas drain line frosted over. This could occur as a result of heat transfer through conduction, but possibly also as a result of the pump fluid from the primary LOX seal drain cavity leaking across the intermediate seal. To prevent a potentially hazardous condition, the purge pressure level in the intermediate seal was

TABLE 2. MARK 48-0 TURBOPUMP TESTING
(P/N RS009820, S/N 01-0)

Test No.	Test Date	Test Duration, Seconds	Accumulated		Remarks
			Starts	Duration, Seconds	
016-011	7-9-76	30	1	30	Initial test using LM ₂ as pumped fluid. Turbine drive media--GH ₂ . 30,800 rpm achieved satisfactorily.
016-012	7-13-76	9	2	39	Pumped fluid: LM ₂ , turbine drive media: GH ₂ targeted rpm: 60,000. Premature cutoff by turbine radial accelerometer vibration safety cutoff system (VSC) exceeded 10 g rms. RPH attained: 45,979.
016-013	7-13-76	5	3	44	Pumped fluid: LM ₂ , turbine drive media: GH ₂ satisfactory rotordynamic test. Maximum turbopump rpm: 61,965.
016-014	7-16-76	70	4	114	Pumped fluid: LOX, turbine drive media: GH ₂ planned H-Q at 30,000 and 60,000 rpm. H-Q obtained at 30,000 rpm. Turbine radial accelerometer VSC cutoff at 15 g rms at 52,500 rpm.
016-015	7-16-76	30	5	144	Pumped fluid: LOX, turbine drive media: GH ₂ planned objective: H-Q mapping at 60,000 rpm. Some H-Q data obtained at 60,850 rpm, but test prematurely cut off by observer due to a fire in a facility system.
016-016	7-16-76	12	6	156	Pumped fluid: LOX, turbine drive media: GH ₂ planned objective: H-Q at 60,000 rpm. Premature cutoff by turbine radial accelerometer VSC system at 52,000 rpm.
016-017	7-16-76	37	7	193	Pumped fluid: LOX, turbine drive media: GH ₂ planned objective: H-Q at 60,000 and 70,000 rpm. Achieved satisfactory H-Q data at 60,000 rpm. Attempted to increase turbopump speed to 70,000 rpm, but was prematurely cut off by turbine radial accelerometer VSC system at a speed of 64,000 rpm. This test concluded series 1 testing. The turbopump and facility system were modified for hot-fire testing with the gas generator system.

TABLE 2. (Continued)

Test No.	Test Date	Test Duration, Seconds	Accumulated		Remarks
			Starts	Duration, Seconds	
016-018	8-3-76	0	8	193	Gas generator ignition not achieved. Cutoff by ignition detect system. Posttest analysis showed problem to be associated with exciter system. Exciter changed prior to next test. Scheduled 60,000 rpm.
016-019	8-3-76	2.81	9	195.81	Objective: 60,000 rpm. Satisfactory test. A turbopump rpm of 57,629 was achieved with a turbine inlet total pressure of 1837 psia at 1761 R (Note: turbine discharge orificing resulted in a turbine pressure ratio of 1.85.) Gas generator c ^a efficiency: 98.9%
016-020	8-3-76	0.58	10	196.39	Objective: 60,000 rpm. Test prematurely terminated by turbine inlet overtemp. Maximum rpm achieved 58,378. Analysis revealed the fuel injection pressure lower than actual controller set pressure. Result: Higher GG mixture ratio with cutoff at 1960 R. Turbine inlet temperature controller readjusted using site data.
016-021	8-3-76	16.58	11	212.97	Objective: 60,000 rpm for test stand duration. Objective partially achieved. Maximum rpm achieved was 52,800, but the test was terminated prematurely by turbine inlet overtemp. Review of data shows main fuel valve position operating in high-flow gain region, only 2-1/2% open. For next test, the H ₂ tank pressure will be reduced to force MFV further open. Gas generator c ^a efficiency: 99.3%
016-022	8-9-76	0	12	212.97	Objective: 70,000 rpm for test stand duration (~50 seconds) Test prematurely terminated by turbine radial accelerometer VSC system. Test terminated during fuel-lead stage at 56,000 rpm and 15 g rms.
016-023	8-9-76	0.62	13	213.59	Objective: 70,000 rpm for test stand duration (~50 seconds) Test prematurely terminated by turbine radial VSC system at 20 g rms. Maximum rpm achieved was 68,725.

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TABLE 2. (Concluded)

Test No.	Test Date	Test Duration, Seconds	Accumulated		Remarks
			Starts	Duration, Seconds	
016-024	8-9-76	1.2	14	214.79	Objective: 70,000 rpm for test stand duration (~50 seconds) Test prematurely terminated by turbine radial accelerometer at 20 g rms. Maximum turbopump rpm: 69,157.
016-025	8-9-76	2.39	15	217.18	Objective: 70,000 rpm for test stand duration (~50 seconds) Test prematurely terminated by turbine inlet overtemp. Fuel injection pressure again below controller set pressure resulting in high GG mixture ratio and overtemp. Maximum rpm: 68,199.
016-026	8-11-76	5.82	16	223.0	Objective: 70,000 rpm for test stand duration (~50 seconds) Test prematurely terminated by observer due to a fire in a facility system. Maximum rpm achieved: 74,191.
016-027	8-11-76	3.01	17	226.01	Objective: 70,000 rpm for test stand duration (~50 seconds) Test prematurely terminated by turbine inlet overtemp. Data analysis showed the fuel injection pressure controller to be lower than required by 69 N/cm ² (100 psi). A site data correction was made for the next test. Maximum rpm achieved: 62,867.
016-028	8-11-76	40.79	18	266.8	Objective: H-Q excursion at 70,000 rpm, and test stand duration (~50 seconds) All objectives except duration were achieved. Manual control of turbopump discharge throttle valve achieved H-Q excursions. Maximum rpm achieved was 68,685. The test was automatically terminated when the intermediate seal purge supply level decreased below 150 psig (redline). The gas generator c* efficiency during the test averaged 99.7%.

raised to 138 N/cm^2 (200 psig). No problem was experienced at this pressure level with mixing of incompatible fluids. It is quite possible that the originally planned purge pressure of 41 N/cm^2 (60 psig) would be adequate. This could be established on future tests by sampling and analyzing the drain fluids during chilldown.

The turbopump was disassembled after the test series to permit visual inspection of the components. Figure 3 shows the condition of the more significant parts. The condition of most of the components was excellent; only two discrepancies were apparent: The pump-end bearings showed evidence of overheating, and the chrome plating on the rotor under the primary hot-gas seal ring had flaked off.

Pump Hydrodynamic Performance. Figure 4 is a plot of the pump overall head rise as a function of flow, where both data and the predicted head are scaled to a speed of 7329 rad/s (70,000 rpm). The scaling was accomplished using the affinity laws which have been thoroughly substantiated as applicable for LOX and LN_2 . The data consist of 66 data points from 15 tests, with test speeds varying from 1628 rad/s (15,550 rpm) to 7768 rad/s (74,190 rpm), and with pumped fluids of both LOX and LN_2 , primarily the former. The symbols used for the data points distinguish the different operating speed ranges tested. There was no indication that the results were dependent on the pumped fluid medium.

The low-speed data show fairly good agreement with the predicted head rise, but may be indicating a slightly steeper H-Q slope than predicted. However, as speed increases, the test data deviate more from the predicted curve, falling short of the curve at the higher flowrates. This type of deviation is typical of that experienced when cavitation is limiting the performance. To investigate this deviation, the ratio ($R_{\Delta H}$) of the test head rise divided by the predicted head rise was calculated and plotted as a function of suction specific speed (N_{SS}) in Fig. 5. The initial plot tended to indicate a great deal of data scatter without clear trend. However, when different symbols were used to represent the different inlet flow coefficients (ϕ_{in}) tested, the data showed a clear trend. For all coefficients, there is a tendency of the head ratio to drop as N_{SS} increases. However, as flow coefficient increases, this dropoff occurs at successively lower values of N_{SS} . This trend again is strongly indicative of cavitation limitations, with the amount of cavitation increasing with either increasing N_{SS} or with increasing flow coefficient at a constant value of N_{SS} .

The cavitation appears to occur at much lower values of N_{SS} than would be expected from the design, considering it does have an inducer designed for good suction performance. This would indicate the more likely possibility that the impeller is cavitating rather than the inducer. This could be caused by:

1. A failure of the inducer to produce its design head rise, which is required to keep the impeller out of cavitation

TABLE 2. (Concluded)

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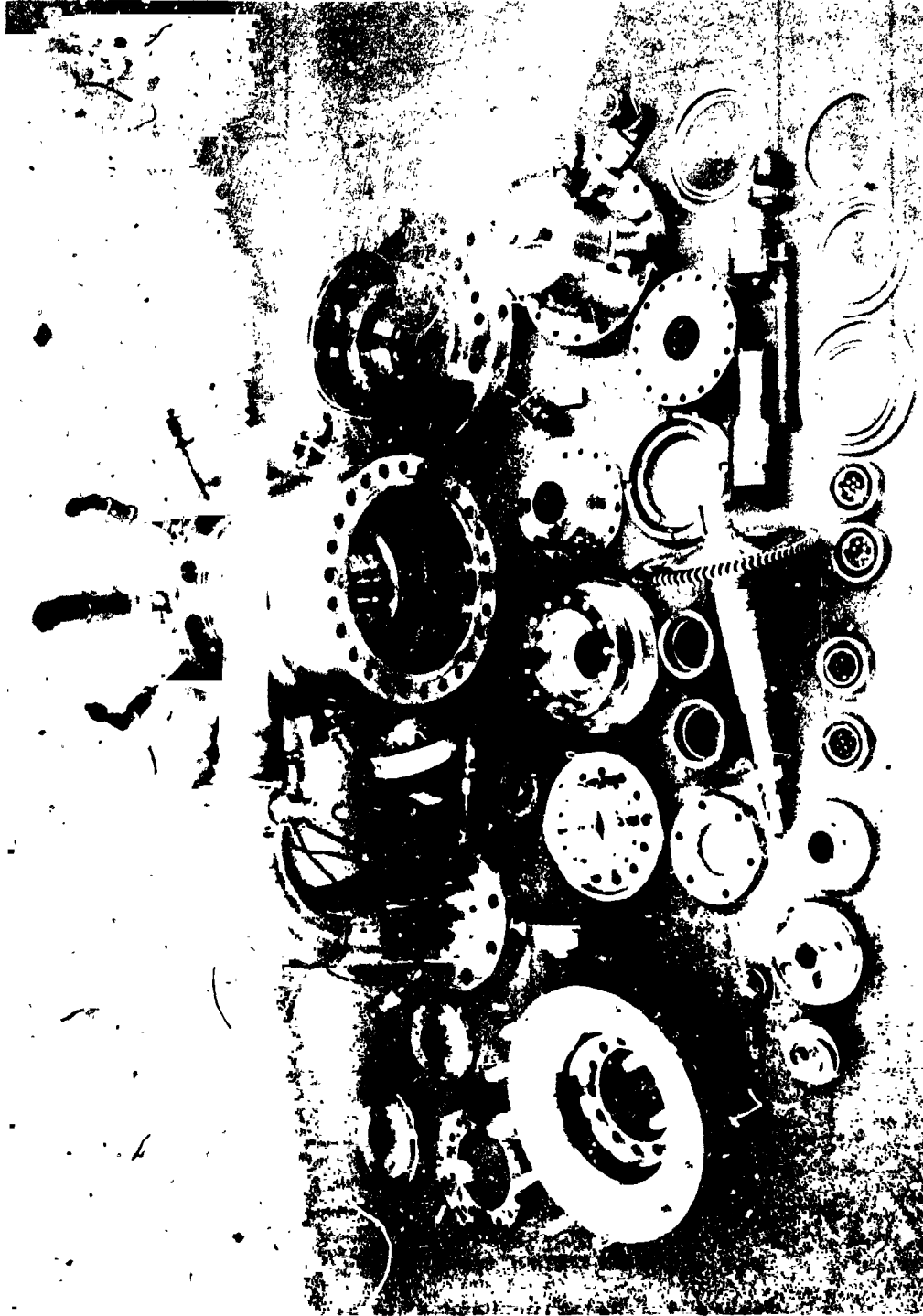
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Figure 3. Mark 48-0 Components After Testing

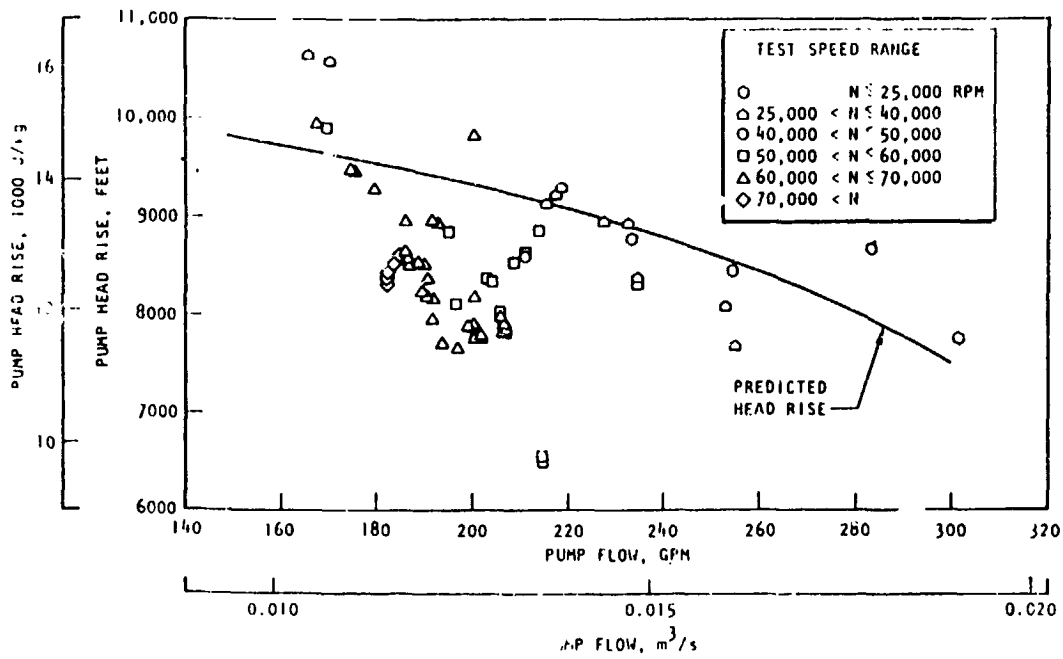


Figure 4. MK 48 LOX Pump Data and Predicted Head Rise Scaled to 70,000 rpm

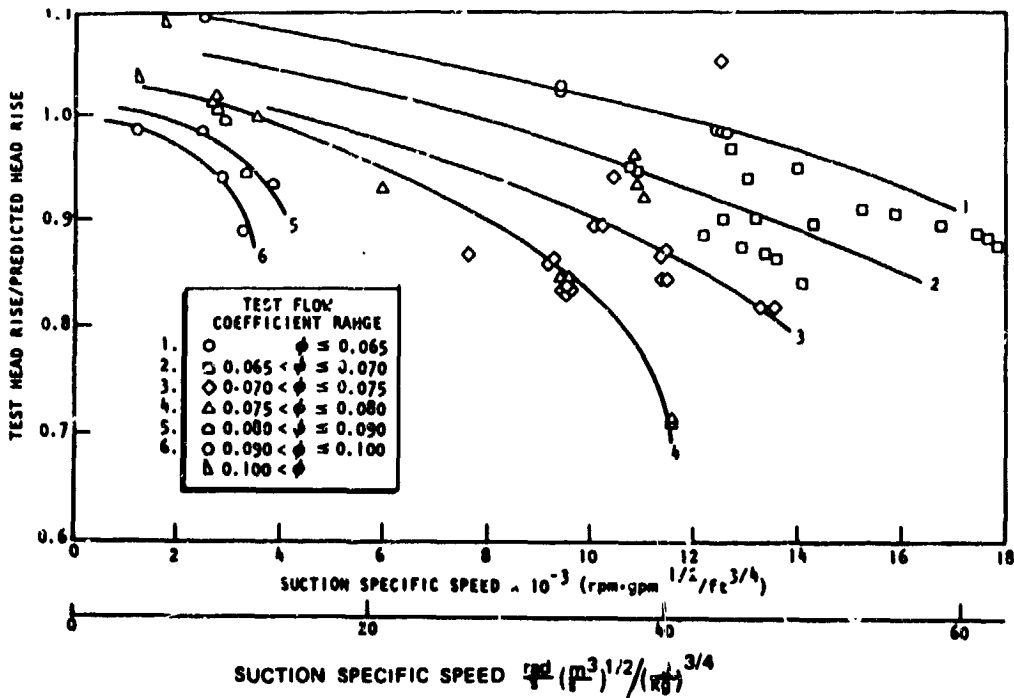


Figure 5. Relative Head Rise as a Function of N_{ss}

2. An inadequate impeller design from a cavitation standpoint
3. Too much hot cryogenic being pumped into the impeller eye from the balance piston/bearing area

Axial Thrust Control. Data from this test series showed the balance piston to be operating in a satisfactory manner, particularly on those tests where part of the flow was bled overboard and, thereby, the return cavity pressure was reduced. To improve the margin in an internal recirculation mode, the size of the return flow passages should be enlarged.

Bearing Coolant Flow. Examination of the bearings post-test showed that the bearings had been overheated. There are two possible explanations:

1. The bearing was overheated during LN₂ tests;
2. The bearing was overheated during the LO₂ tests.

These two possibilities are distinguished because experience with bearings in LN₂ operation. Rocketdyne's experience in this fluid medium has been inconsistent, some tests indicating satisfactory operation, others showing definite signs of bearing distress. The bearings from the Mark 48 had a very similar appearance to others damaged during LN₂ operation. Because of this earlier experience, the total test time in LN₂ was purposely kept to a minimum; three tests were conducted with a total duration of 44 seconds and a maximum rotor speed of 6492 rad/s (62,000 rpm).

Regardless of the LN₂ operation, however, there are indications that the LOX flow through the bearings could be substantially less than was desired, and that the temperature of the coolant was potentially higher than expected. The data have already been used to show that the balance piston thrust range in some cases was less than the design range. This limitation was attributed to the higher resistance downstream of the balance piston sump. This same high resistance tends to restrict the coolant flow.

The overheating condition would be made worse by the possible larger loads carried by the bearing due to the inability of the balance piston to develop the thrust range desired in some instances. The load tracks on the bearings were wider than usual, indicating variable loading conditions.

The third factor affecting the bearing temperatures is the temperature of the coolant fluid itself in and around the bearings. Figure 6 shows the temperature in the balance piston return flow area as a function of speed. Many of the temperatures experienced are actually warmer than any encountered previously with LOX bearings. It is desirable to keep the temperature down to approximately 110 K (200 R). Data in Fig. 6 show temperatures as high as 160 K (290 R) at speeds of 6282 rad/s (60,000 rpm). The higher temperatures noted on the earlier tests were a cause of concern that led to the action of opening an instrumentation line as an overboard bleed of the balance piston flow return cavity. This change was made effective on Test 19 and subsequent and, even though the return port was small, the data of Fig. 6 show that there

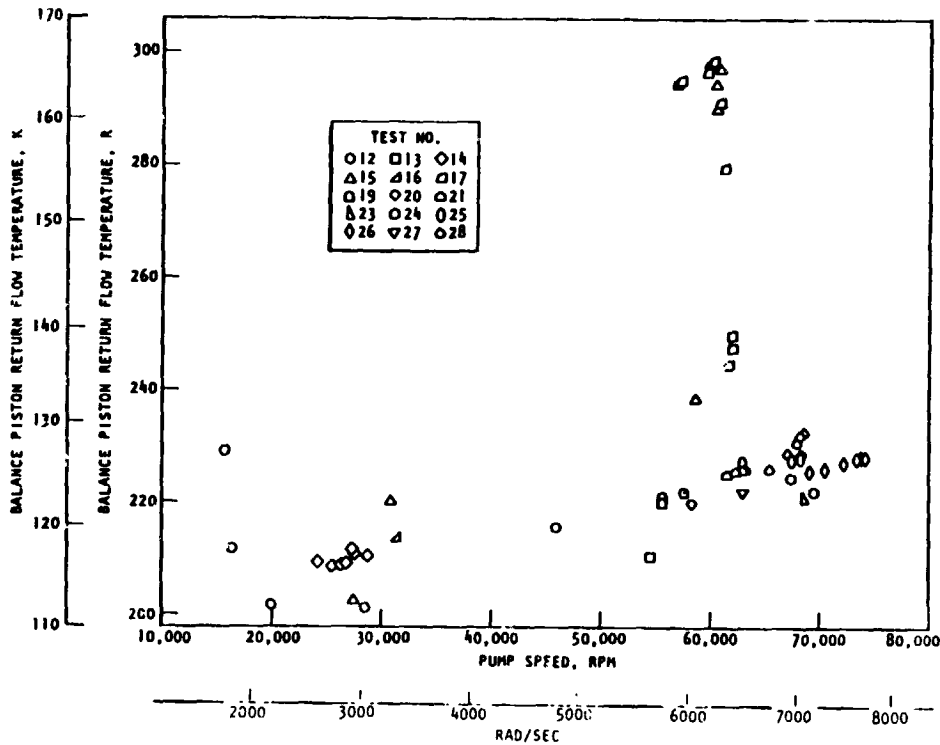


Figure 6. Mark 48-0 Turbopump Speed vs Balance Piston Return Flow Temperature

was definite tendency to lower the temperature in this cavity. Subsequent tests were able to get to speeds of 7330 rad/s (70,000 rpm) or higher without exceeding approximately 130 K (235 R). Thus, the overheating initially must be at least partially due to insufficient coolant flowrate out of this cavity area. This same problem leads to a higher back pressure at the balance piston sump, and results in the lower thrust range previously reported.

A need for further analyses to explore the coolant flow problem is indicated. These analyses can be expected to cover the effects of the heating due to power disk drag on the back side of the impeller and on the slinger. Preliminary analyses indicate that, at a 7330 rad/s (70,000 rpm), the impeller back side power disk drag could easily result in a temperature increase of 17 K (30 R), with the flows calculated in analyzing the balance piston performance.

ANALYSIS AND DESIGN MODIFICATIONS

Hydrodynamic Analysis

The purpose of the initial phase of this effort was to analyze in-depth the hydrodynamic characteristics evidenced in the data obtained during the test series conducted in July and August 1976. Based on the results of the analysis, design changes were made for incorporation into the next turbopump

to be tested. The purpose of the modifications was to provide additional instrumentation, correct obvious deficiencies, and reroute the balance piston flow in an external recirculation mode where the flow can be measured and the amount recirculated varied.

Suction Performance Analysis. The data obtained during the initial test series of the turbopump revealed that the head produced by the pump dropped off at increased suction specific speeds and flow coefficients. This tendency pointed toward a cavitation problem either in the inducer or impeller. Therefore, effort was initiated to re-examine the inducer hydrodynamic design and performance analytically. The Dynatech computer program was utilized for this purpose. The independent check indicated no problem with the inducer performance. The discharge pressures calculated using the Dynatech program approached originally predicted values closely, and in some instance exceeded them. The predicted head-flow characteristics are included in Fig. 7. Based on the results of this analysis no further action was taken with regard to the inducer. To verify the analytical performance, a pressure measurement was included between the inducer and impeller.

Analytical effort was also directed at the impeller hydrodynamic design to identify elements which could potentially cause or contribute to the poor suction performance of the pump. The static pressure distribution along the flow passage was calculated and it was found that the through flow area near the leading edge of the impeller blade was restricted more than what would

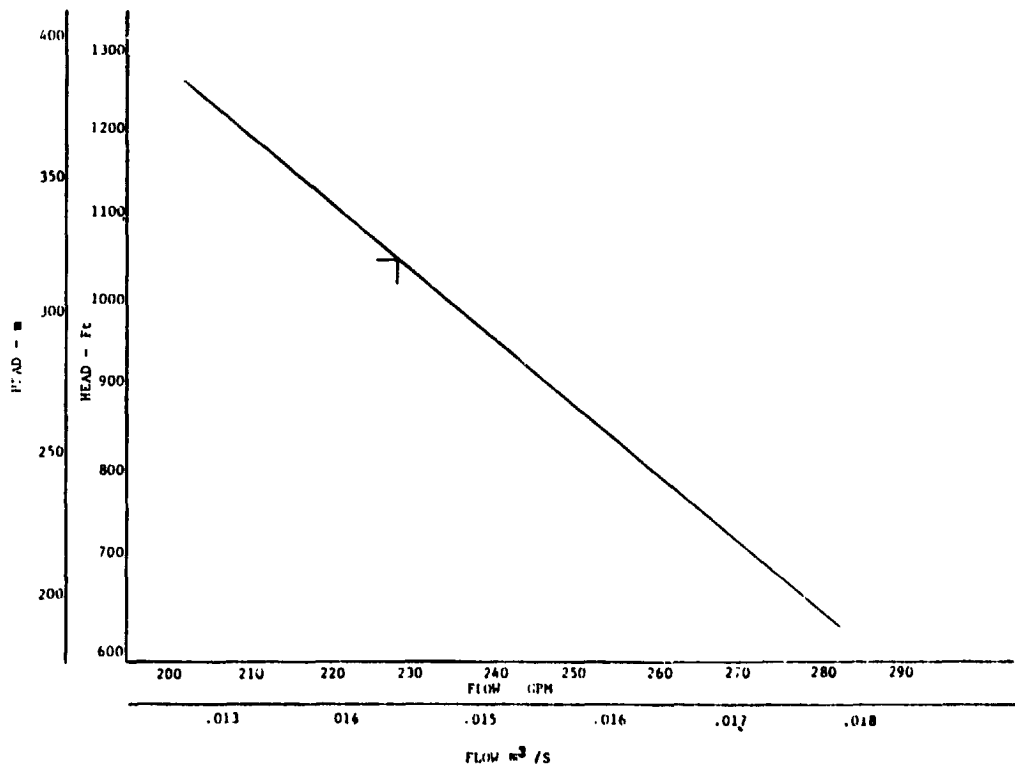


Figure 7. MK 48-0 Inducer Head-Flow Characteristics

be conducive to good hydrodynamic performance. This condition actually causes a negative head rise at the blade inlet of 229m (751 feet). In Table 3 the pertinent parameters of the Mark 48-0 impeller are compared with those of the Mark 44-0 impeller which is of similar size and which has exhibited excellent hydrodynamic characteristics. Even through the pressure level present at the impeller leading edge would appear to be sufficient to preclude local cavitation, it is evident that the NPSH margin at that point is substantially less with the Mark 48-0. Whether actual cavitation is present or not, the overall pressure generating capability of the impeller is reduced by this condition.

One of the postulated reasons for the low head generated by the impeller was the balance piston return flow which is dumped into the eye of the impeller. It was hypothesized that this relatively high-temperature fluid is vaporizing and causing excessive blockage in the impeller. To investigate this theory an analysis was made of the entire balance piston flow loop.

Detailed calculations were made of five test slices in which the fluid state points at various significant stations of the flow circuit were established. The location of these stations are identified in Fig. 8. Actual pressure measurements are available at Stations 1, 2, 4, and 6. The temperature was monitored at Station 6, and it could be derived for Station 1 with a good degree of accuracy from pump inlet and discharge temperature measurements. The pressures and temperatures at the other stations were calculated. The five test slices selected for detailed analysis were typical representatives of the various conditions under which the pump was operated in the initial test series. A summary of the selected data slices is included in Table 4.

Table 5 presents the statepoints for Test No. 014, time slice 8, which was a low-speed run 2739 rad/s (26,157 rpm) in which all of the balance piston fluid was recirculated internally into the eye of the impeller. The data of Table 5 is plotted on a Mollier diagram for LOX in Fig. 9. As can be seen both from the density values in Table 5 and the location of the points in Fig. 9, the fluid remains liquid throughout the flow process.

TABLE 3. COMPARISON OF MK 48-0 AND MK 44-0 IMPELLERS

	MK 48-0	MK 44-0
(1) Impeller Head, m(ft)	3658 (12,000)	889 (2916)
(2) Inducer Head Rise, m(ft)	305 (1,000)	123 (600)
(3) Leading Edge Static Head Increase, m (ft)	-229 (-751)	230 (+755)
(4) IMP Leading Edge Static Head Rise Above Pump Inlet, m (ft)	75.9 (249)	413 (1355)
(5) IMP Leading Edge Head Rise Impeller Head Rise	0.0208	0.464

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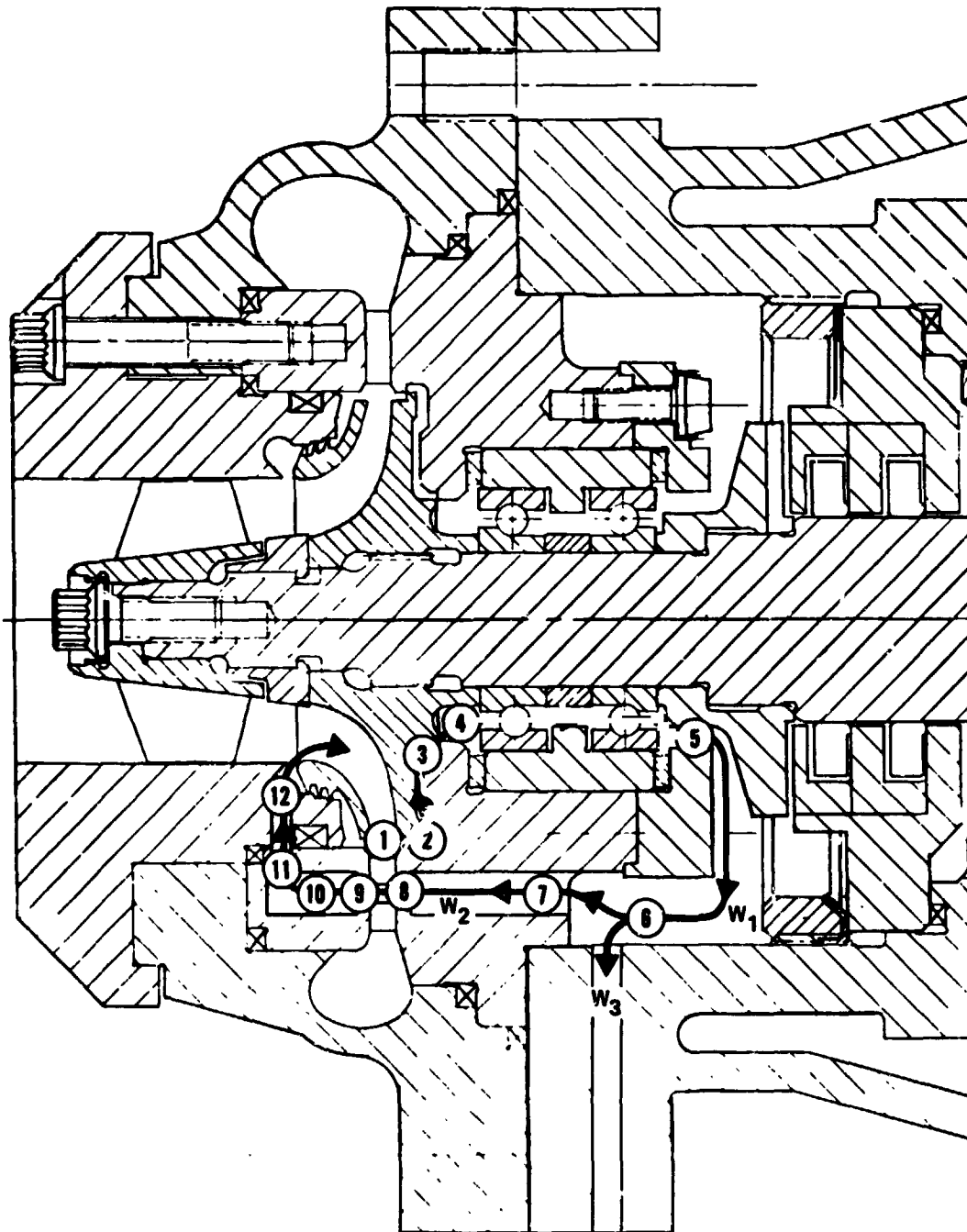


Figure 8. MK 40-8 Balance Piston Flow State Points

TABLE 4. SELECTED TEST POINTS FOR BALANCE PISTON FLOW ANALYSIS

Data Slice	Speed, rad/s (rpm)	Balance Piston Flow Routing
014-8	2738 (26,157)	All internal recirculation
017-8	5980 (57,110)	All internal recirculation
019-3	5997 (57,629)	Partial overboard bleed
024-1	7242 (69,157)	Partial overboard bleed
025-3	7142 (68,199)	Partial overboard bleed

TABLE 5. MK 48-0 BALANCE PISTON FLOW STATE POINTS, TEST NO. 014-8

<p>Speed = 2739.2 rad/s (26,157 rpm) $W_1 = 0.466 \text{ kg/s (1.028 lb/sec)}$ $W_2 = 0.466 \text{ kg/s (1.028 lb/sec)}$ $W_3 = 0.0 \text{ kg/s (0.0 lb/sec)}$</p>			
Station	Pressure, N/CM ² (psia)	Temperature, K (R)	Density, gm/cc (lb _m /ft ³)
1	480.6 (697.1)	98.9 (178.1)	1.1065 (69.07)
2	444.2 (644.2)	99.0 (178.2)	1.1049 (68.97)
3	389.2 (564.5)	99.5 (179.1)	1.1007 (68.71)
4	345.4 (500.9)	99.6 (179.2)	1.0996 (68.64)
5	331.2 (480.3)	99.7 (179.5)	1.0979 (68.53)
6	362.7 (526.1)	116.1 (208.9)	0.9932 (62.0)
7	362.7 (526.1)	116.1 (208.9)	0.9932 (62.0)
8	308.3 (447.1)	116.1 (208.9)	0.9932 (62.0)
9	283.1 (410.6)	116.1 (208.9)	0.9932 (62.0)
10	179.4 (260.2)	116.1 (208.9)	0.9932 (62.0)
11	178.5 (258.9)	116.1 (208.9)	0.9932 (62.0)
12	175.0 (253.8)	116.1 (208.9)	0.9932 (62.0)

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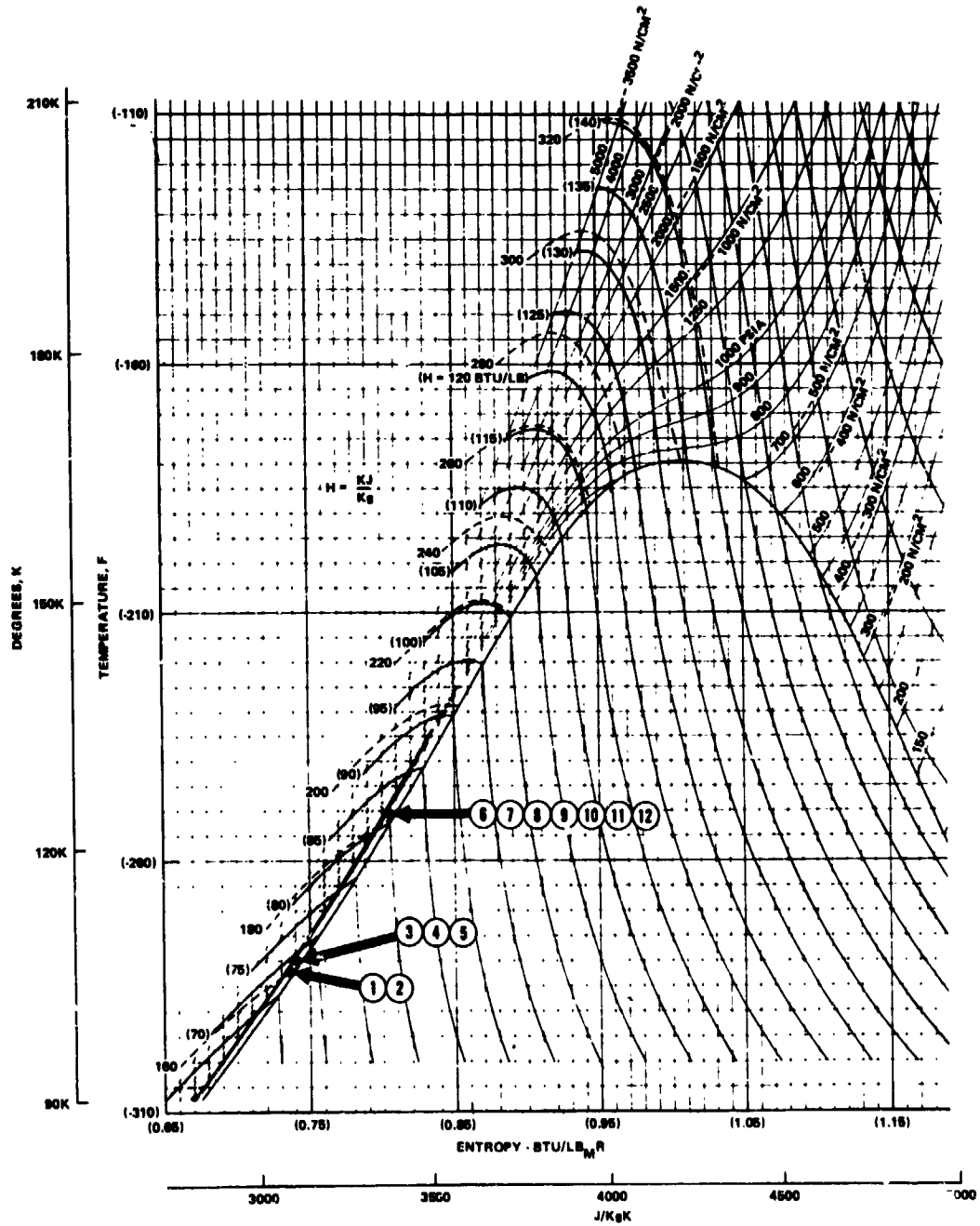


Figure 9. MK 48-0 Balance Piston Fluid State Points

The results of a similar analysis are given in Table 6 and Fig. 10 for Test No. 017-8 in which all of the balance piston fluid was still recirculated internally, but the operating speed has higher 5980 rad/s (57,110 rpm). It can be seen that in this case a substantial vaporization takes place, and a relatively low density fluid is injected into the inlet of the impeller.

As a next step in the analysis three test slices were examined in which about half of the balance piston flow was bled overboard. The results are shown in Tables 7, 8, 9, and on the Mollier diagram in Fig. 11. The first data slice (019-3) is at essentially the same operating speed as Test No. 017-8 previously examined for the case of full-internal recirculation, while the two subsequent data slices are near the design speed of 7330 rad/s (70,000 rpm). It is evident that with partial overboard bleed the fluid recirculated into the impeller eye does not vaporize.

The principal cause for vapor formation in the high-speed, full-recirculation mode was the high temperature reached by the fluid at Station 6. It was postulated that the return flow passage area through the diffuser vanes was inadequate, causing a high return cavity pressure at Station 6 which in turn reduced the flow through the bearings. If the heat input from impeller fluid viscous shear effects, bearing power loss and the slinger power input remained constant, the fluid at a lower flowrate would be raised to a higher temperature.

TABLE 6. MK 48-0 BALANCE PISTON FLOW STATE POINTS, TEST NO. 017-8

Speed = 5980 rad/s (57,110 rpm)			
$W_1 = 0.7114 \text{ kg/s (1.5707 lb/sec)}$			
$W_2 = 0.7114 \text{ kg/s (1.5707 lb/sec)}$			
$W_3 = 0.0 \text{ kg/s (0.0 lb/sec)}$			
Station	Pressure, N/CM ² (psia)	Temperature, K (R)	Density, gm/cc (lb _m /ft ³)
1	1440.3 (2088.9)	109.6 (197.3)	1.077 (67.24)
2	1214.1 (1760.8)	110.1 (198.1)	1.069 (66.75)
3	1059.1 (1536.1)	117.1 (210.7)	1.041 (65.0)
4	986.2 (1430.3)	117.1 (210.7)	1.033 (64.5)
5	945.5 (1371.3)	117.6 (211.7)	1.033 (64.5)
6	953.9 (1383.5)	163.6 (294.4)	0.673 (42.0)
7	953.8 (1383.3)	163.6 (294.4)	0.673 (42.0)
8	766.0 (1111.0)	160.1 (288.2)	0.609 (38.0)
9	670.3 (972.1)	158.4 (285.2)	0.625 (39.0)
10	285.3 (413.8)	140.0 (252.0)	0.232 (14.5)
11	276.5 (401.0)	139.7 (251.5)	0.229 (14.3)
12	259.1 (375.8)	139.4 (251.0)	0.224 (14.0)

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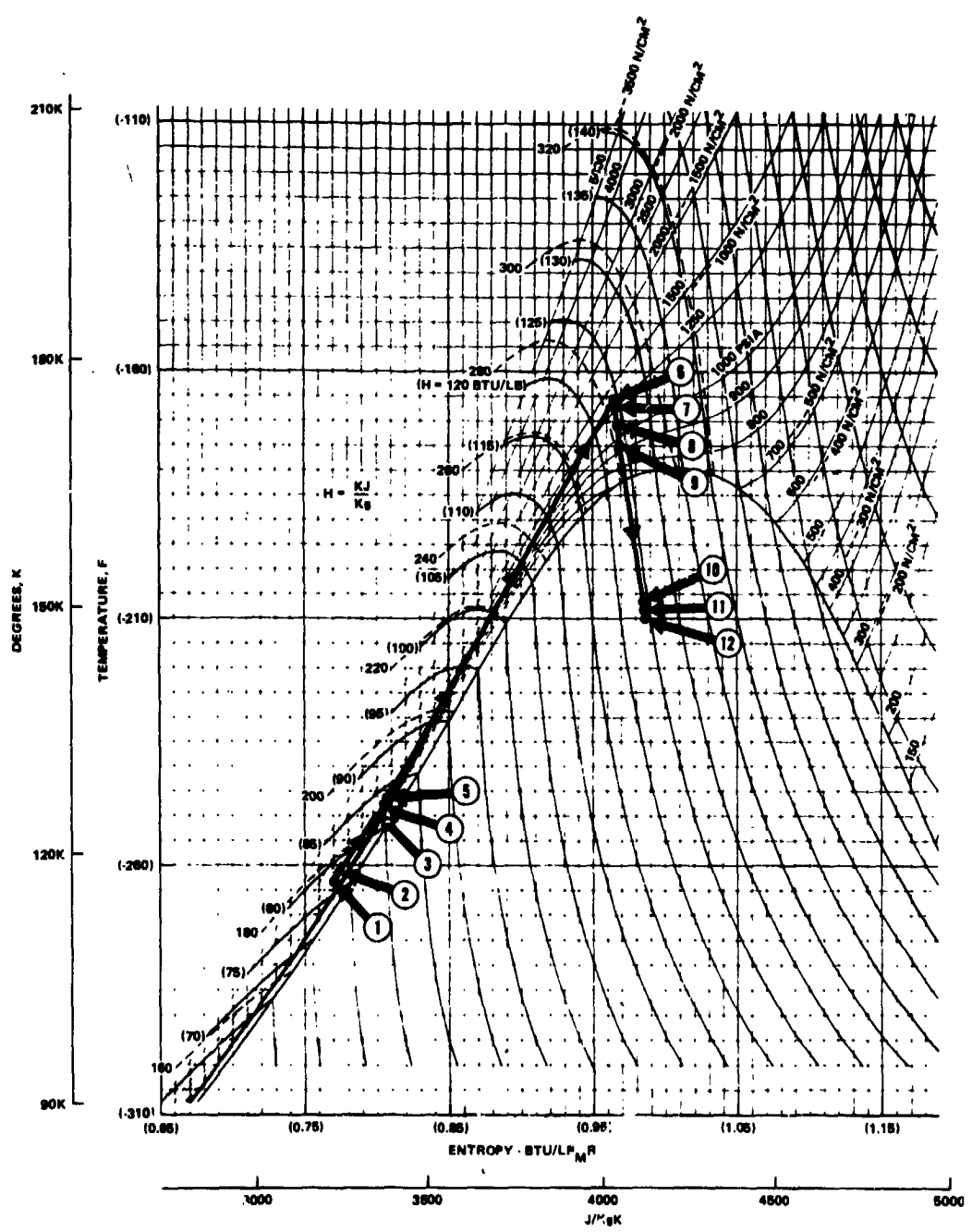


Figure 10. MK 48-0 Balance Piston Fluid State Points
(Internal Recirculation Test No. 017-8)

TABLE 7. MK 48-0 BALANCE PISTON FLOW STATE POINTS, TEST NO. 019-3

Speed = 6034.9 rad/s (57,629 rpm) $W_1 = 1.246$ kg/s (2.745 lb/sec) $W_2 = 0.672$ kg/s (1.484 lb/sec) $W_3 = 0.571$ kg/s (1.261 lb/sec)			
Station	Pressure, N/CM ² (psia)	Temperature, K (R)	Density, gm/cc (lb _m /ft ³)
1	595.8 (2314.5)	106.7 (192.0)	1.097 (68.45)
2	1192.7 (1729.8)	107.9 (194.2)	1.078 (67.3)
3	1008.8 (1463.0)	109.5 (197.1)	1.067 (66.58)
4	743.4 (1078.2)	110.2 (198.3)	1.055 (65.85)
5	646.7 (937.9)	110.3 (198.6)	1.050 (65.57)
6	659.6 (956.6)	123.0 (221.4)	0.977 (61.0)
7	659.5 (956.5)	123.0 (221.4)	0.977 (61.0)
8	554.2 (789.2)	123.0 (221.4)	0.977 (61.0)
9	490.9 (711.9)	123.0 (221.4)	0.977 (61.0)
10	271.2 (393.3)	123.0 (221.4)	0.977 (61.0)
11	269.3 (390.6)	123.0 (221.4)	0.977 (61.0)
12	252.3 (365.9)	123.0 (221.4)	0.977 (61.0)

TABLE 8. MK 48-0 BALANCE PISTON FLOW STATE POINTS. TEST NO. 024-1

Speed = 7242 rad/sec (69,157 rpm) $W_1 = 1.211$ kg/s (2.673 lb/sec) $W_2 = 0.610$ kg/s (1.347 lb/sec) $W_3 = 0.601$ kg/s (1.326 lb/sec)			
Station	Pressure, N/CM ² (psia)	Temperature, K (R)	Density, gm/cc (lb _m /ft ³)
1	2019.1 (2928.4)	11.0 (198.0)	1.094 (68.3)
2	1501.0 (2177.0)	111.1 (200.0)	1.075 (67.1)
3	1010.0 (1464.8)	112.2 (201.9)	1.049 (65.5)
4	780.4 (1131.8)	112.8 (203.0)	1.041 (65.0)
5	680.5 (986.9)	113.6 (204.5)	1.033 (64.5)
6	680.5 (987.0)	123.2 (221.8)	0.977 (61.0)
7	680.5 (986.9)	123.2 (221.8)	0.977 (61.0)
8	585.3 (848.9)	123.2 (221.8)	0.977 (61.0)
9	541.5 (785.3)	123.2 (221.8)	0.977 (61.0)
10	360.3 (522.6)	123.2 (221.8)	0.977 (61.0)
11	358.8 (520.4)	123.2 (221.8)	0.977 (61.0)
12	318.2 (461.5)	123.2 (221.8)	0.977 (61.0)

TABLE 9. MK 48-0 BALANCE PISTON FLOW STATE POINTS, TEST NO. 025-3

Speed = 7141.8 rad/s (68,199 rpm) $W_1 = 1.447 \text{ kg/s (3.195 lb/sec)}$ $W_2 = 0.781 \text{ kg/s (1.725 lb/sec)}$ $W_3 = 0.666 \text{ kg/s (1.470 lb/sec)}$			
Station	Pressure, N/CM ² (psia)	Temperature, K (R)	Density, gm/cc (lb _m /ft ³)
1	2138.1 (3100.9)	109.9 (197.9)	1.097 (68.5)
2	1542.7 (2237.4)	110.9 (199.7)	1.075 (67.1)
3	1360.2 (1972.7)	113.1 (203.5)	1.057 (66.0)
4	1006.5 (1459.7)	114.3 (205.7)	1.041 (65.0)
5	873.5 (1266.8)	114.4 (206.0)	1.025 (64.0)
6	873.9 (1267.4)	126.5 (227.7)	1.096 (60.0)
7	837.8 (1267.3)	126.5 (227.7)	1.096 (60.0)
8	715.3 (1037.4)	126.5 (227.7)	1.096 (60.0)
9	642.2 (931.4)	126.5 (227.7)	1.096 (60.0)
10	340.5 (493.8)	126.5 (227.7)	1.096 (60.0)
11	337.9 (490.1)	126.5 (227.7)	1.096 (60.0)
12	301.1 (436.7)	126.5 (227.7)	1.096 (60.0)

The conclusion that the fluid returned to the impeller eye did not vaporize on tests which used a partial overboard bleed had a significant impact on the predicted suction capability of the pump. It was originally assumed that the low-suction performance obtained during the initial test series was attributable in a large measure to two-phase fluid being ingested into the impeller eye from the balance piston loop; and it was expected that by improving the condition of the balance piston fluid, a substantial improvement in the pump suction capability would be realized. But since the generated head was low on those data slices on which the returned fluid was in a liquid state, it was concluded that a modification of the pumping elements was required to effect an improvement in the suction performance.

Balance Piston Fluid Temperature Analysis. Temperature measurements of the balance piston fluid in the return cavity, downstream of the bearings and the slinger, were higher than predicted in general, and substantially higher during the tests on which the fluid was fully recirculated. To investigate the cause for the higher temperatures, the heat input to the fluid by the slinger ribs was calculated for two data slices: 017-8, on which the fluid was recirculated; and 025-3 on which the balance piston fluid was partially bled overboard.

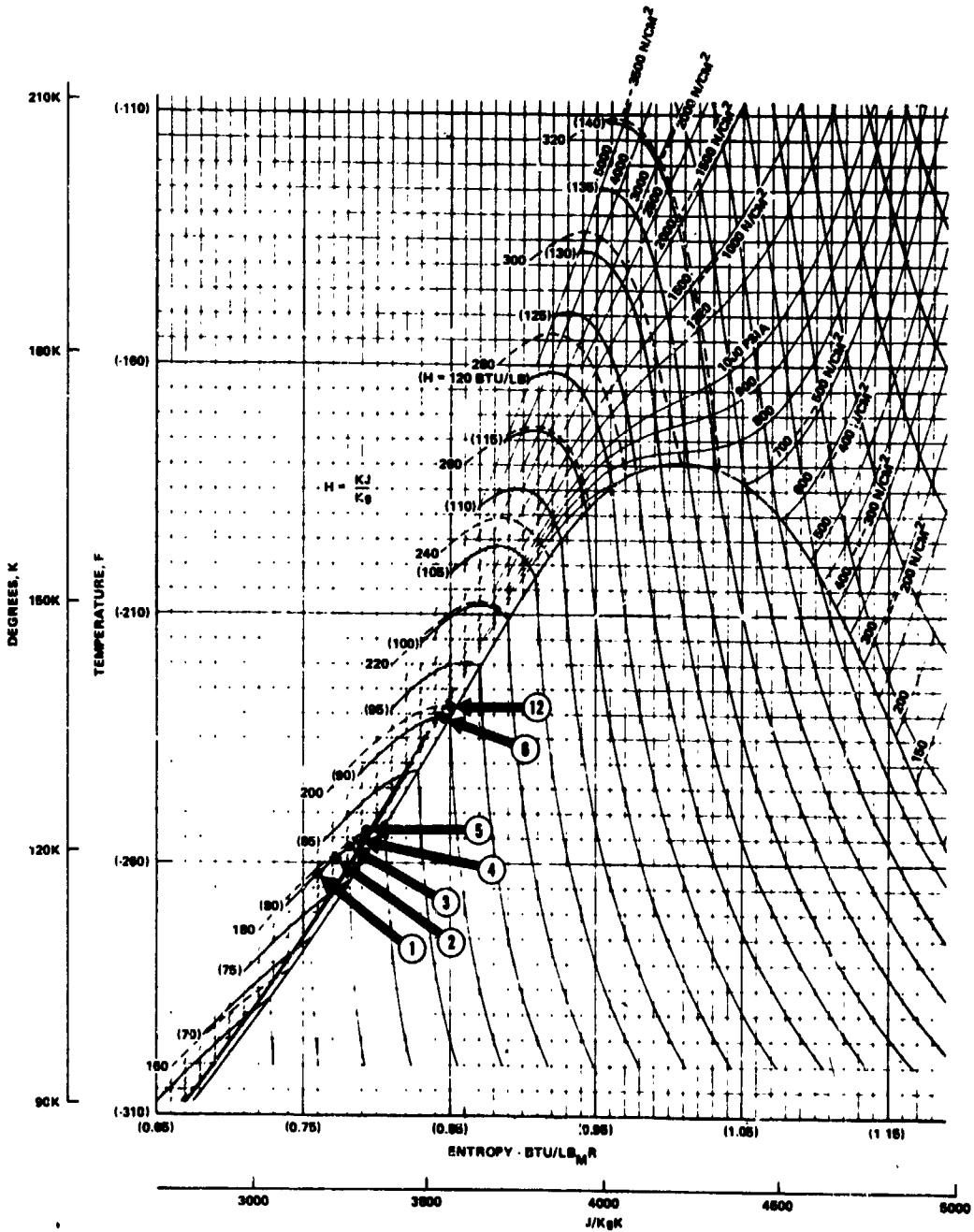


Figure 11. MK 48-0 Balance Piston Fluid State Points With Overboard Bleed (Test No. 025-3)

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The calculated pressure profile as a function of position along the slinger radius is shown in Fig. 12 for test slice 017-8. The 163 K (294 R) fluid temperature measured in the return cavity, vapor pressure is reached at a radius of 2.41 cm (0.95 inch). In Fig. 13 the temperature rise attributable to the slinger as a function of the radius at which vaporization occurs is presented for a range of coefficients. As can be seen from the figure, if the fluid vapor interface is at 2.41 cm (0.95 inch), as predicted for Test 017-8, the contribution of the slinger to the temperature rise is calculated in the range 2.5 to 3.9 K (4.5 to 7.0 R). In contrast, the actual measured temperature was 44 K (80 R) higher than the predicted temperature in the cavity. It was concluded from the above that for this test slice the high recorded temperature was not caused by a high-heat input from the slinger.

A similar analysis was performed for Test 025-3. The radius at which vaporization occurs is established 2.41 cm (0.93 inch) in Fig. 14, and the corresponding slinger-caused temperature rise is in the range 3.96 to 6.1 K (7 to 11 R), as indicated by Fig. 15. In this case, the measured temperature approximates the predicted value within 5.5 R (10 R). Neither of the two cases analyzed indicates a great amount of heat input from the slinger. It should be noted that the slinger was installed with a relatively high axial clearance 2.69 inch (0.106 inch) at the ribs. This could have introduced secondary flow effects which were not taken into account in the above an analysis, and which may have some impact on the heat input.

The investigation relative to the slinger effects was extended to explore the feasibility of removing the slinger altogether from the assembly or reducing its outer diameter. The primary purpose of the slinger is to limit the leakage rate through the LOX seal by reducing the pressure at the seal entrance to below vapor pressure. Therefore, to determine whether the slinger can be moved, the seal leakage rate and its ultimate impact must be assessed. LOX seal leakage rates were calculated with and without a slinger; the obtained values are shown in Table 10.

TABLE 10. CALCULATED LOX SEAL LEAKAGE RATES

With Slinger	
(a) With Normal Balance Piston Return Cavity Temperature (228 R)	0.017 kg/s (0.038 lb/sec)
(b) With High-Balance Piston Return Cavity Temperature (294 R)	0.052 kg/s (0.115 lb/sec)
Without Slinger	0.213 kg/s (0.47 lb/sec)

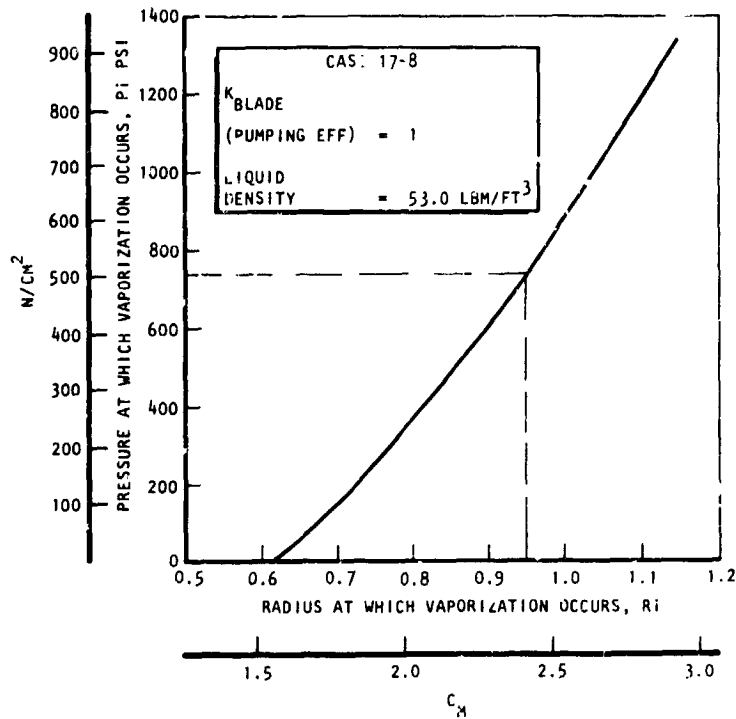


Figure 12. Mark 48-0 Oxidizer--Predicted Pressure Distribution on Backside of Slinger Without Vaporization

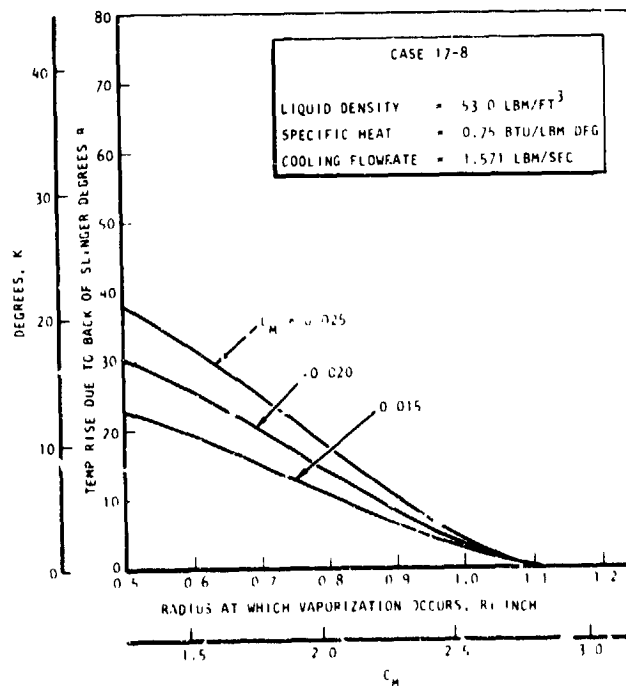


Figure 13. Mark 48-0 Oxidizer--Predicted Cooling Flow Temperature Rise Due to Heat (Torque) Input of Bladed Backside of Slinger

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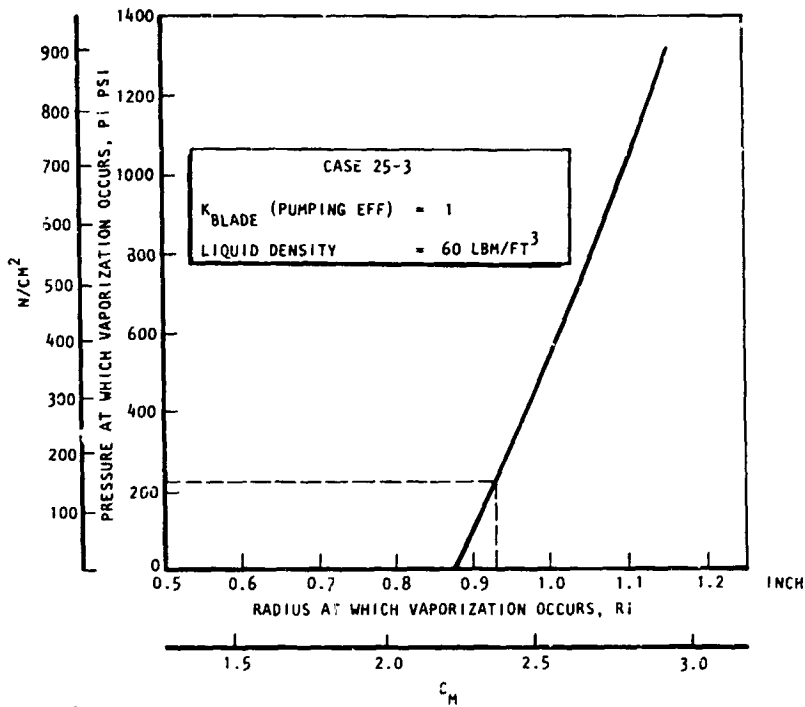


Figure 14. Mark 48-0 Oxidizer--Predicted Pressure Distribution on Backside of Slinger Without Vaporization

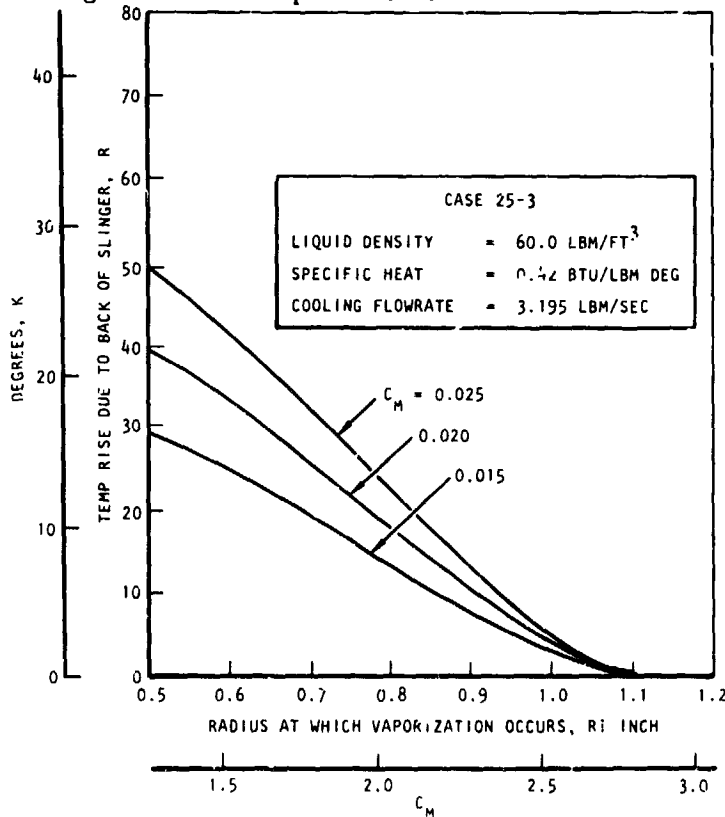


Figure 15. Mark 48-0 Oxidizer--Predicted Cooling Flow Temperature Rise Due to Heat (Torque) Input of Bladed Backside of Slinger

As part of this analysis, the question addressed was could the slinger radius be reduced to some intermediate level where it would add a smaller energy to the fluid, but still maintain the primary seal inlet pressure at the vapor level.

A parametric curve was generated (Fig. 16) in which the required slinger radius to attain vapor at the seal was plotted as a function of balance piston return cavity pressure. The parameter "k" refers to the assumed slinger pumping efficiency; its value is estimated as 0.9.

With a pumping efficiency of 0.9, the slinger radius could have been trimmed to approximately 2.5 cm (1.0 inch), provided the return cavity pressure remained below (1400 psi) as was the case on the last test series in conjunction with a partial overboard bleed. But with an anticipated increase in the general pressure level with the elimination of cavitation, it was expected that the cavity pressure would rise to a higher level. Therefore, to maintain the primary LOX seal leakage at a safe level, the slinger radius was maintained at the original 2.9 cm (1.15 inches) for the ensuing test series.

Design Modifications

Several modifications were made to the turbopump hardware prior to the second test series to incorporate additional instrumentation, and to rectify a deficient condition.

To assess the potential performance improvement to be realized by opening up the impeller inlet area, one impeller was reworked for the test series. The rework consisted of increasing the eye diameter from 4.19 to 4.44 cm (1.65 to 1.75 inches), and cutting back the leading edge of the blades 30 degrees of wrap or approximately 0.95 cm (3/8 inch) at the mean diameter. The leading edges of the cut-back blades were center-line faired. A photograph showing an impeller of the original configuration and one after rework is shown in Fig. 17.

The second test series planned was to be conducted with the balance piston flow routed externally to the impeller inlet or overboard. To accommodate this a design change was made to the main housing to increase the capacity of the overboard drain system from the balance piston return cavity. This was accomplished by converting an existing static pressure port located in the return cavity to an overboard bleed line by increasing its passage diameter. Capability for monitoring pressure in the return cavity was preserved by incorporating a small static pressure line in another section.

To facilitate returning the balance piston flow from an external source to the impeller inlet, a design change was made to the pump inlet housing. The change, illustrated in Fig. 18, adds a circumferential manifold from which LOX is introduced through axial passages to the impeller inlet.

In conjunction with the provisions for routing the balance piston flow externally, the normally internal passage was blocked by pressing dowel pins in the 13 radial passages in the inlet housing.

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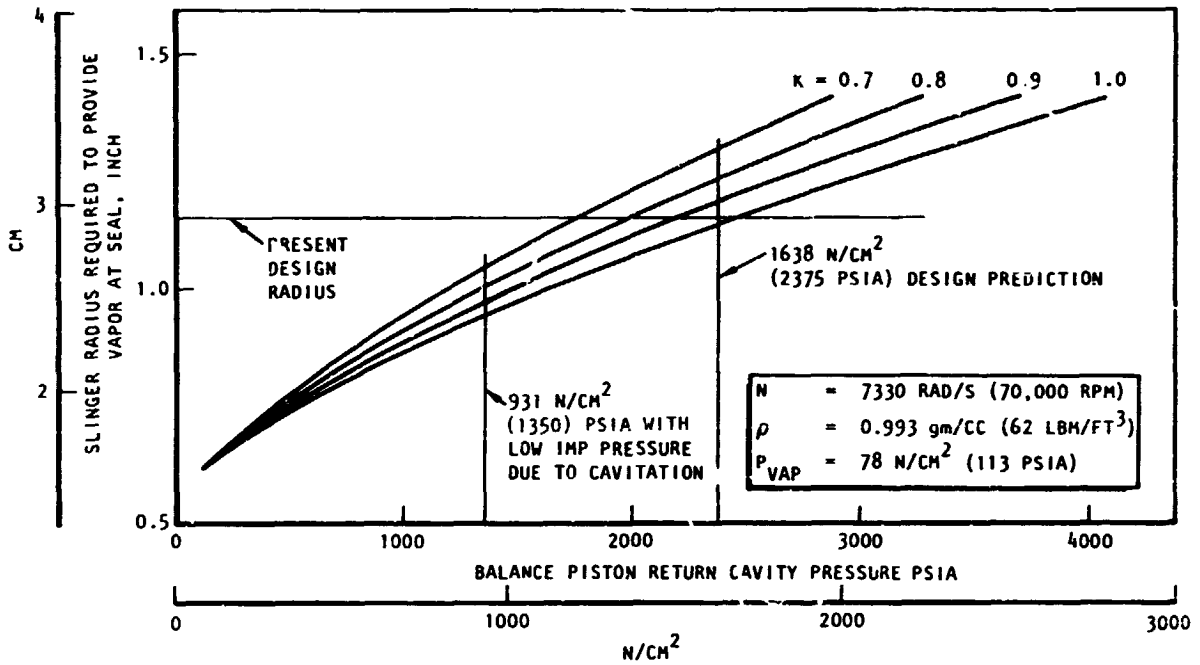


Figure 16. Slinger Required Radius (No Safety Factor) As a Function of Balance Piston Return Cavity Pressure for Various Values of Pumping Efficiency

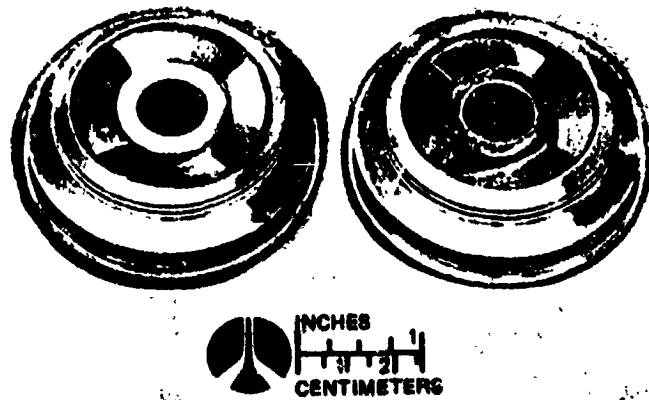
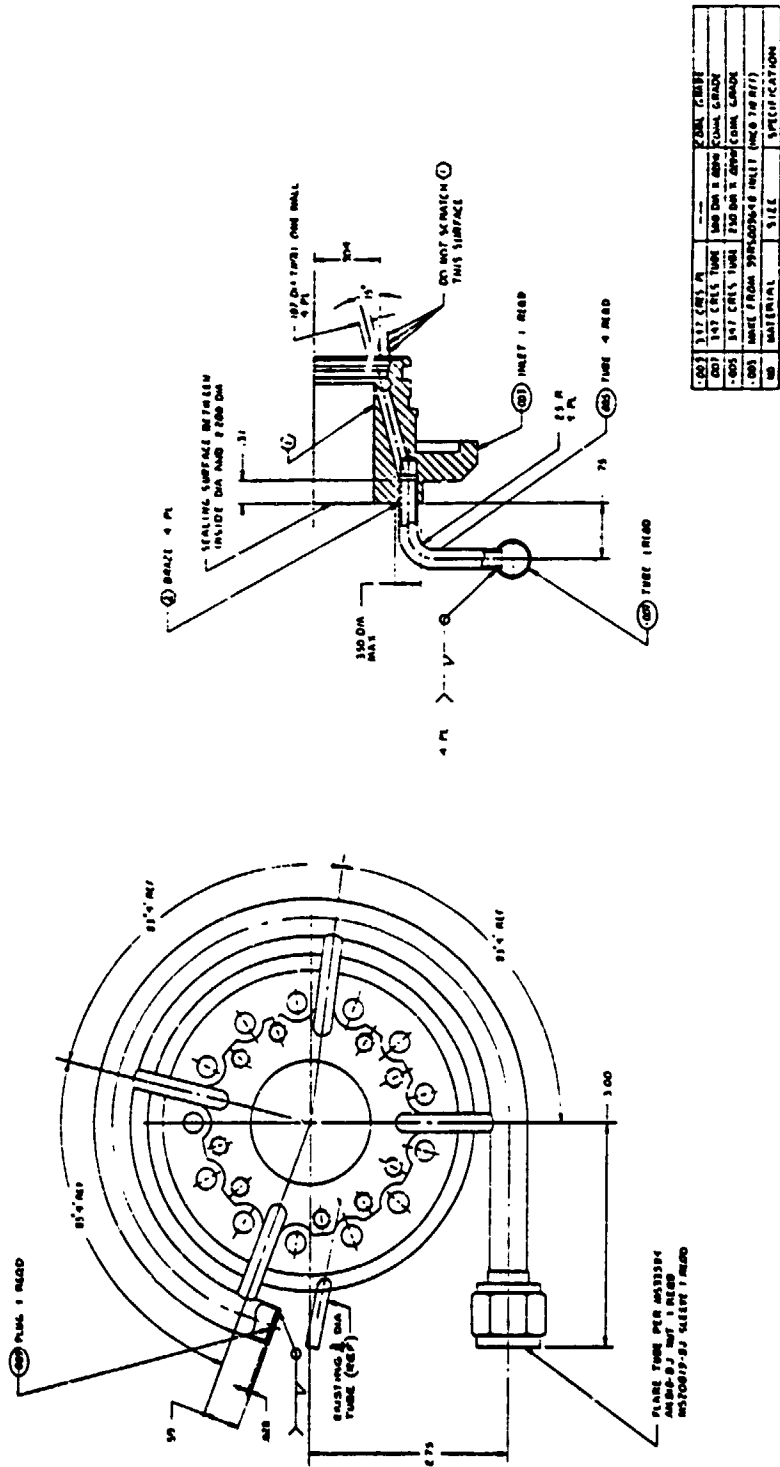


Figure 17. MK 48-0 Reworked Impeller (left) Shown With Unmodified Impeller



CONTACT ALIENOR BIRMAN (P&M) FOR BRASS
 THIS SURFACE IS SILVER PLATED

Figure 18. Internal Housing With Balance Piston Return Manifold Feature

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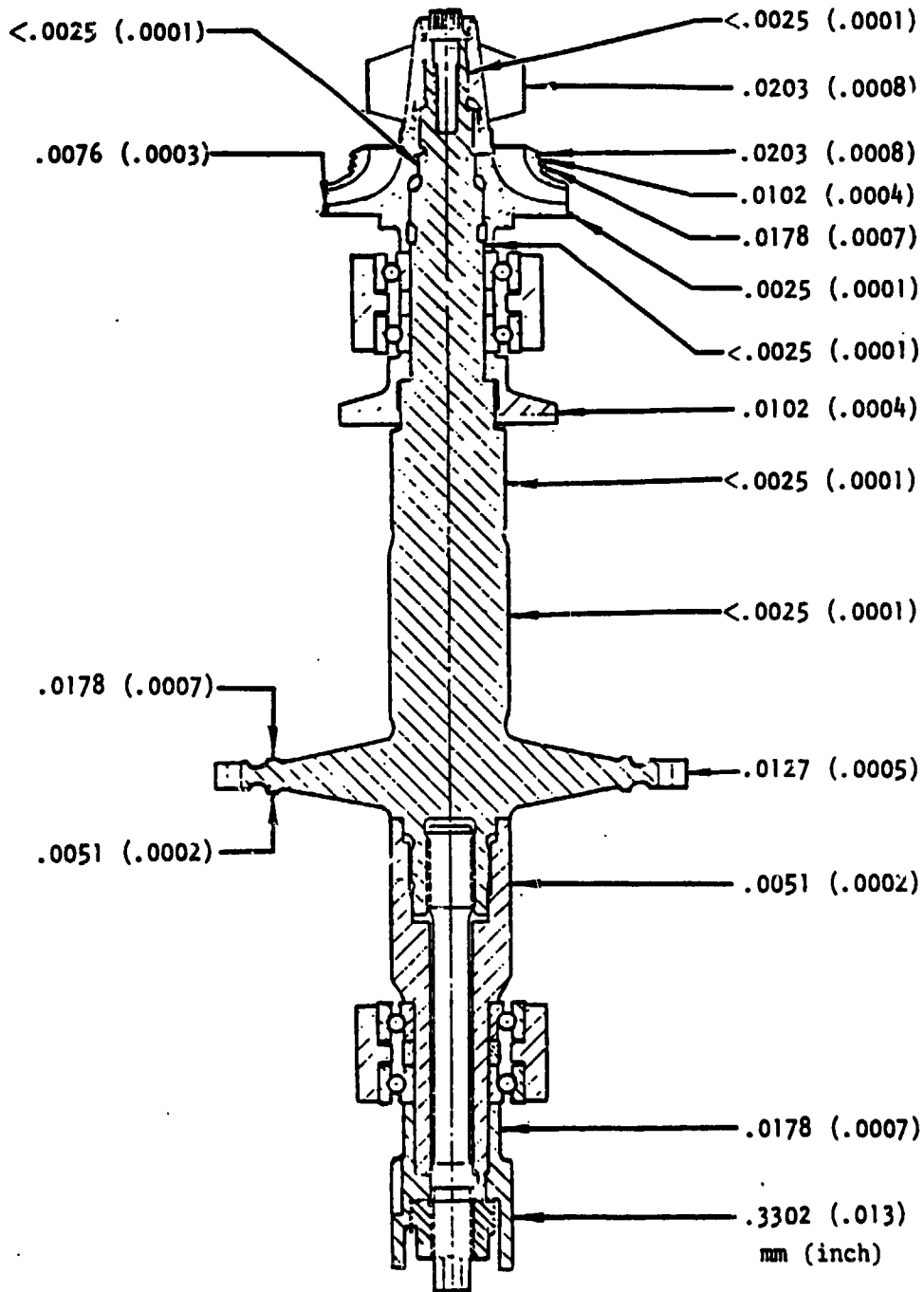


Figure 19. MK 48-0 Turbopump S/N 01-1 Runouts

Modifications of the slinger to reduce the axial clearance between the slinger ribs and the primary LOX seal housing from 2.7 mm (0.106 inch) to approximately 0.176 mm (0.007 inch) were completed. In view of the reduced clearance, a layer of silver plating 0.25 mm (0.010 inch) thick was applied to the primary seal housing to minimize the possibility of a hard metal-to-metal rub.

The inlet housing was changed to include a static pressure measurement between the inducer and impeller. These data were needed to ascertain whether the apparent cavitation at high speeds and high-flow coefficients is caused by insufficient head generated by the inducer or by impeller blockage from the returning balance piston flow.

A static pressure port was also incorporated to the rear bearing support to monitor the turbine wheel downstream pressure. This was required for two purposes: To validate the pressure level applied to the exhaust side of the wheel in the rotor axial thrust calculations; and to assess the magnitude of the pressure drop in the turbine exhaust manifold.

Difficulties encountered in analytically predicting the value of the whirl coefficient on the rear shroud of the impeller, between the high- and low-pressure orifices of the balance piston made it highly desirable to add another pressure tap to that cavity. The original balance piston cavity pressure tap is located approximately in line with the radius of the high-pressure orifice. A second pressure tap was added just outside the radius of the low-pressure orifice.

TURBOPUMP S/N 01-1 ASSEMBLY

Rotor Balance

Dynamic balancing of the MK 48-0 rotor was accomplished on a Gisholt balance machine with a capability for detecting 6×10^{-4} mm (25 μ inch) radial motion was used. For the Mark 48-0 rotor mass of 2.84 kg (6.25 lbs.), this translates into machine accuracy limit of 0.18 gm cm (0.07 gm inch), which would cause a radial load of 98 N (22 lbs.) at the design speed of 7330 rad/s (70,000 rpm). The rotor was supported in the balance cradle by two pairs of turbopump bearings, each pair axially preloaded in the bearing cartridge exactly as in the turbopump assembly. Balancing was initiated using the main rotor and the rear stub shaft assembly, and wax corrections were made in the plane of the turbine wheel and the stub shaft.

Subsequently, the slinger, impeller, inducer, and instrumentation sleeves were added, making wax correction in the plane of each component before the next part was added. After the wax corrections were completed, several repeatability checks were made in which the rotor was disassembled and reassembled, and the change in residual imbalance was established, and the runouts at several stations were measured. Satisfactory repeatability was obtained in the amounts of imbalance as well as the runouts of the parts. The final runout values are shown in Fig. 19. Subsequently, the permanent balance of the rotor was effected by grinding material in designated areas of the component parts.

Turbopump Assembly

The assembly of Turbopump S/N 01-1 was accomplished in accordance with the procedure described in Ref. 1. The front and rear bearing inner race thicknesses were selected to provide a devised minimum bearing preload of 245 N (55 lbs), and to obtain a total bearing travel within each cartridge of 0.23 mm (0.009 inch). The resulting preload curves for the forward (pump end) and aft bearing set are illustrated in Fig. 20 and 21.

Measurements were made during the assembly of the turbopump to establish critical clearances and fits. The diametral fits obtained relative to the bearings are noted in Fig. 22. Critical clearances in the pump, shaft seals, and turbine area are included in Fig. 23 through 25.

After the turbopump was assembled, a push-pull test was performed on the rotor to establish the external loads which the bearings support as a function of rotor position with respect to the balance piston orifice positions. The curve which was obtained is shown in Fig. 26. The symbols h_1 and h_2 refer to the balance piston high- and low-pressure orifice axial clearances, respectively. As indicated by the curve, the bearing stops were positioned such that the balance piston orifices would overlap (i.e., h_1 and h_2 would be negative) by 0.0047 inch and 0.0046 inch, respectively, before a sizeable load (450 lbs) would be imposed on the bearings.

Test Series No. 2

The purpose of the test series was to define the hydrodynamic performance of the pump with and without balance piston fluid recirculation. The initial test was planned with all of the balance piston flow routed overboard. Head-flow characterization tests at 3141 rad/s (30,000 rpm) and 7330 rad/s (70,000 rpm) were planned, with a suction performance test at 7330 rad/s (70,000 rpm) and nominal pump flow.

The second test was planned to determine the effect of recirculating the balance piston flow. After obtaining a reference point at 7330 rad/s (70,000 rpm), and nominal flow, approximately half of the balance piston fluid was planned to be recirculated into the impeller eye. This was to be followed by a noncavitating data point at nominal pump flow, with all of the fluid recirculated, and subsequently flowing by a cavitation data point.

A third test series was tentatively planned in which suction performance at off design flow was to be explored, budget and schedule permitting.

Installation. The turbopump was installed in Lima test cell at the Propulsion Research Area of Rocketdyne's Santa Susana Field Laboratory. A simplified schematic of the facility is shown in Fig. 27.

In Fig. 28, a detail schematic of the balance piston recirculation and overboard flow system is presented. Flow control and measurement orifices as well as remote control valves were included in each leg of the system to facilitate varying the amount of flow recirculated from zero to 100%.

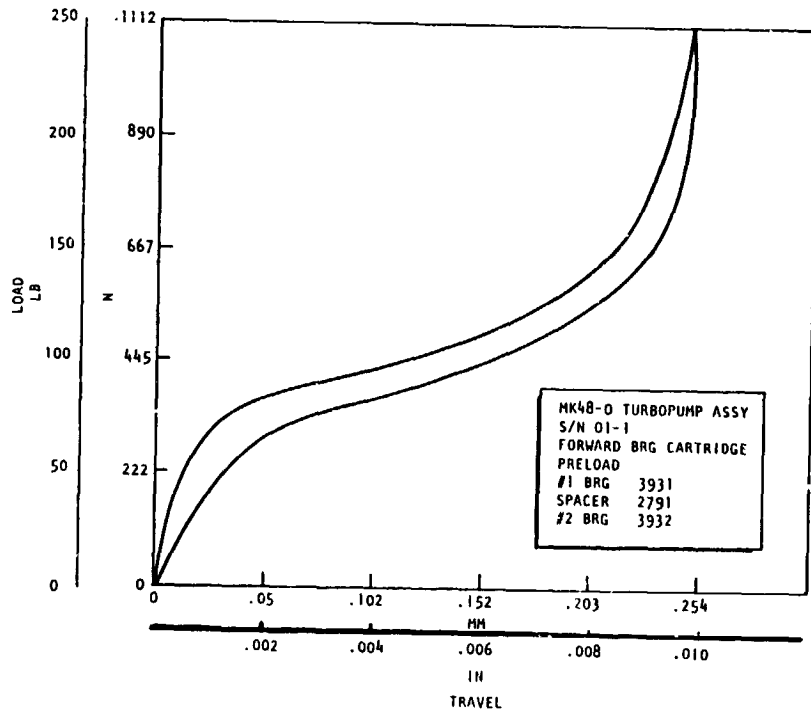


Figure 20. Mark 48-G Turbopump Assembly S/N 01-1 Forward Bearings Cartridge Preload

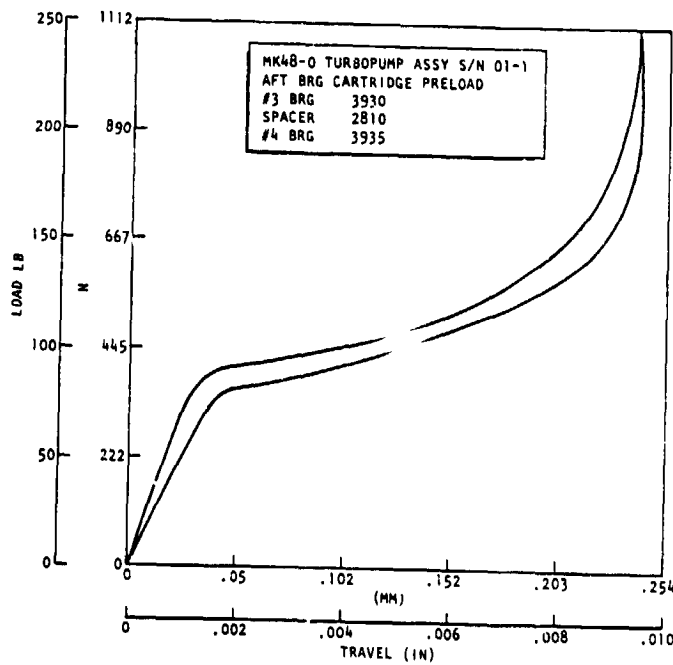


Figure 21. Mark 48-0 Turbopump Assembly S/N 01-1 Aft Bearings Cartridge Preload

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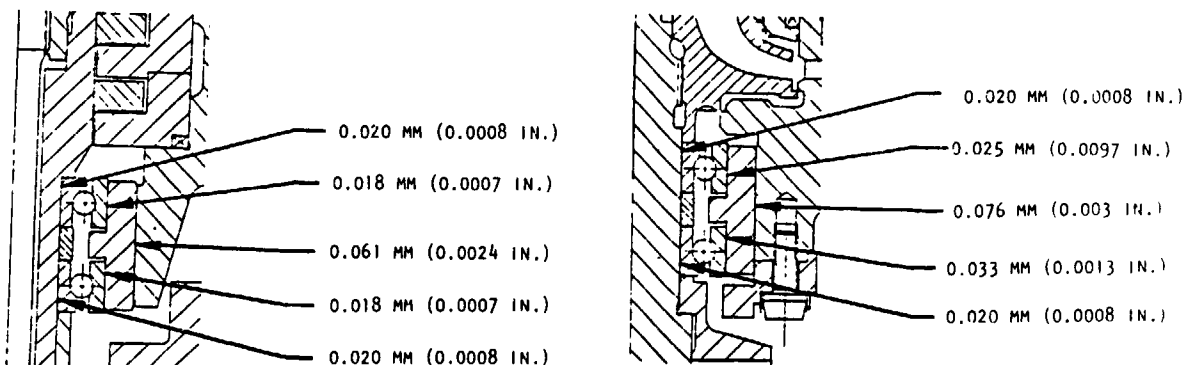


Figure 22. Mark 48-0 Turbopump S/N 01-1 Bearing Fits

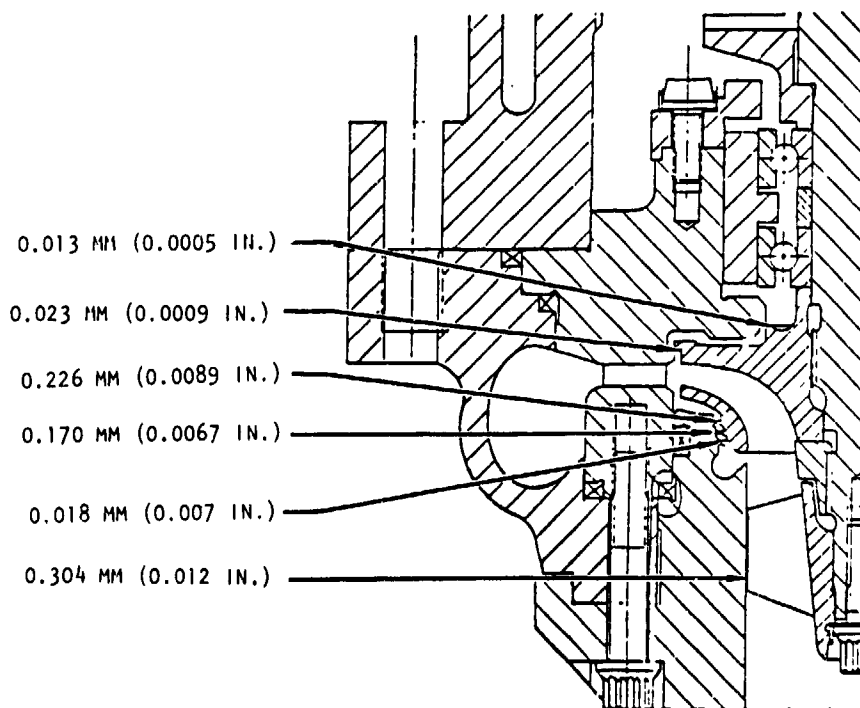


Figure 23. Mark 48-0 Pump Diametral Clearances

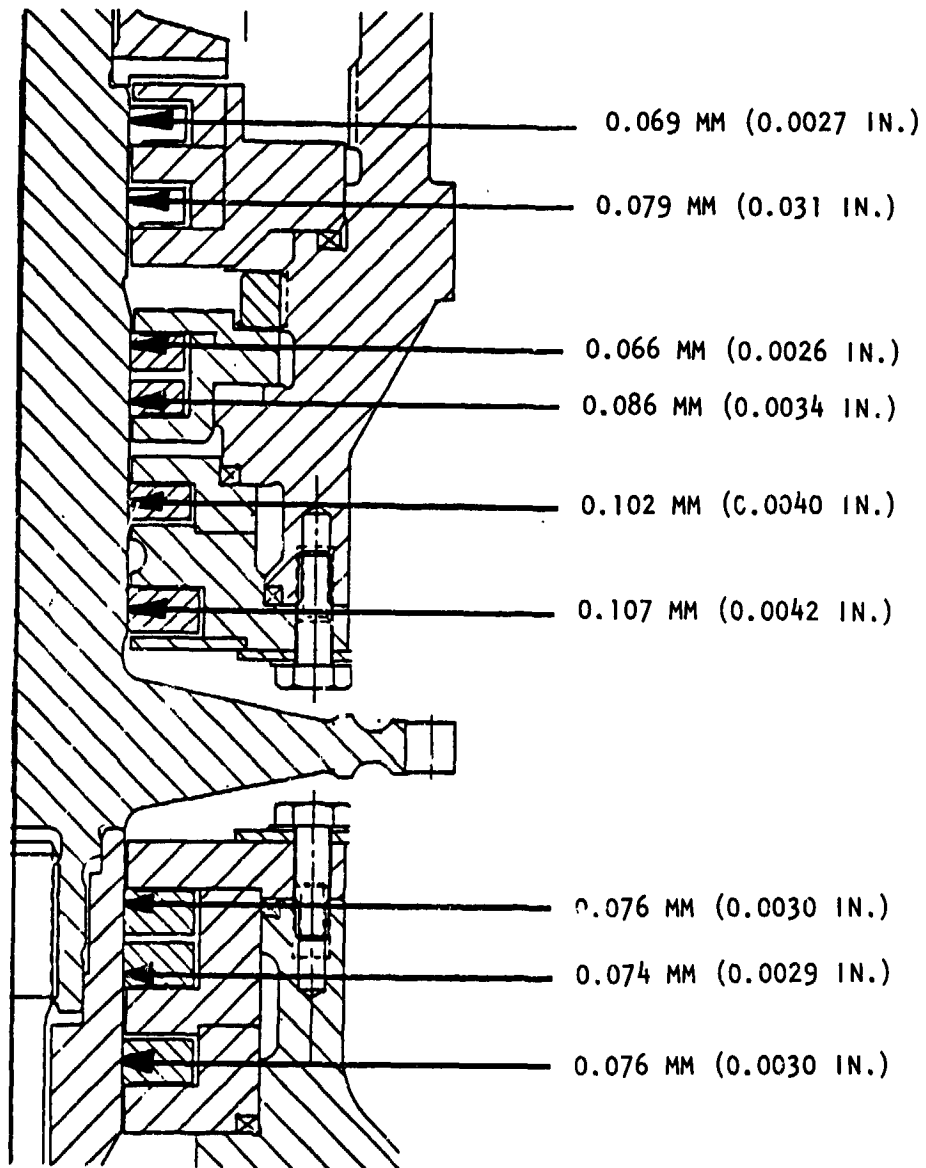


Figure 24. MK 48-0 S/N 01-1 Seal Diametral Clearances

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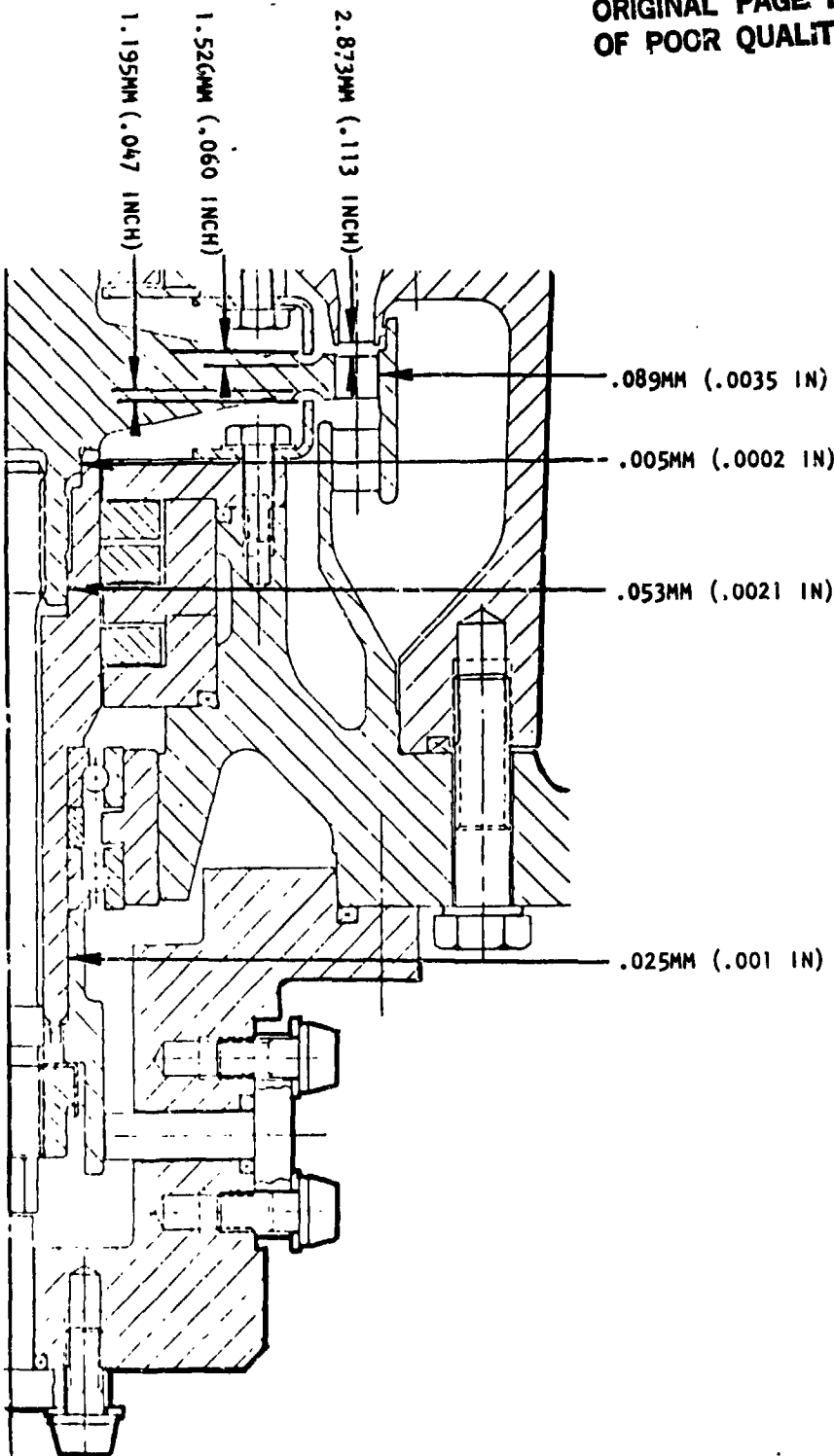


Figure 25. MK 48-0 Turbopump S/N 01-1 Turbine End Clearances and Fits

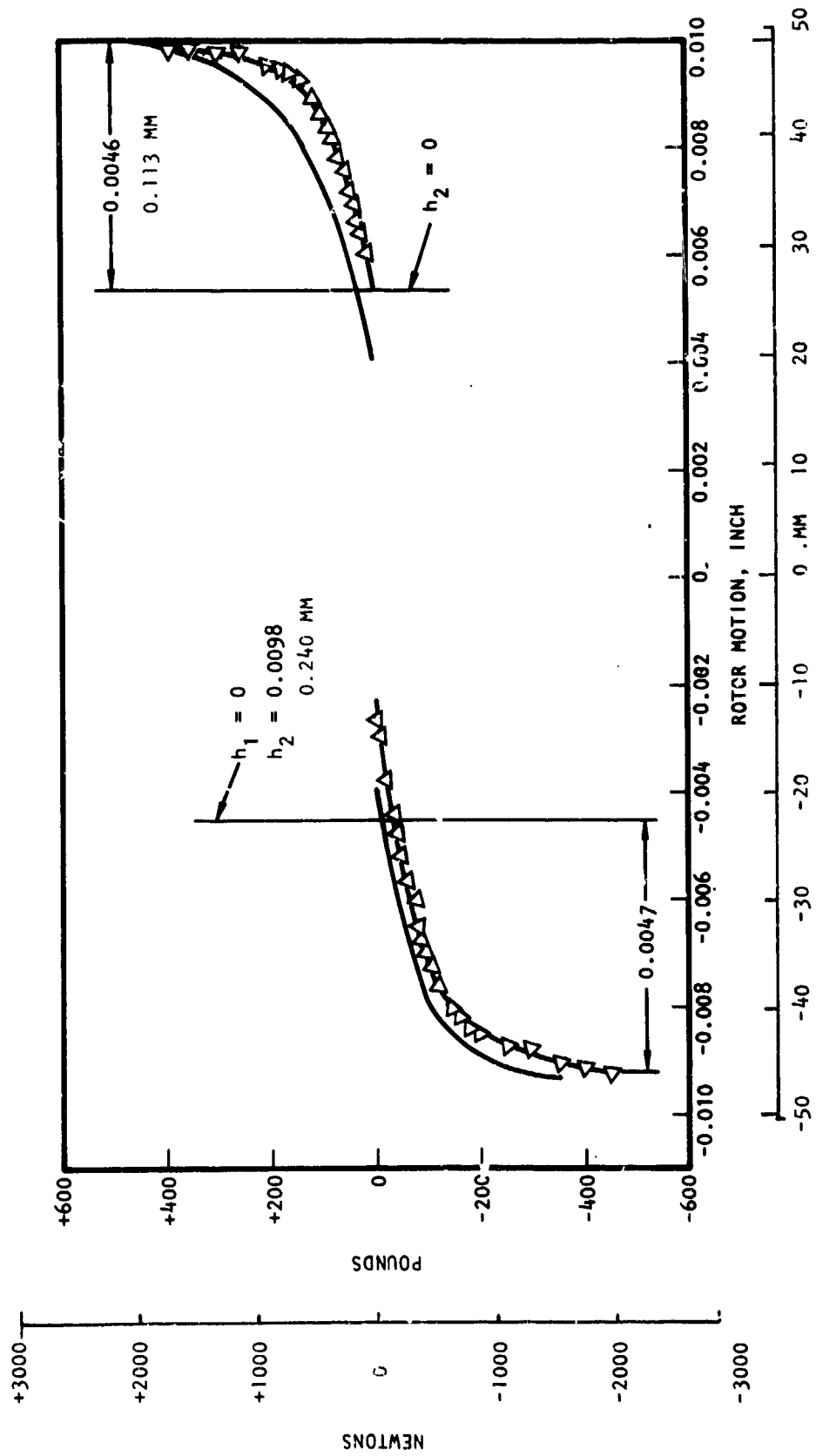


Figure 26. Mark 48-0 S/N 01-1 Final With Turbopump Rotor

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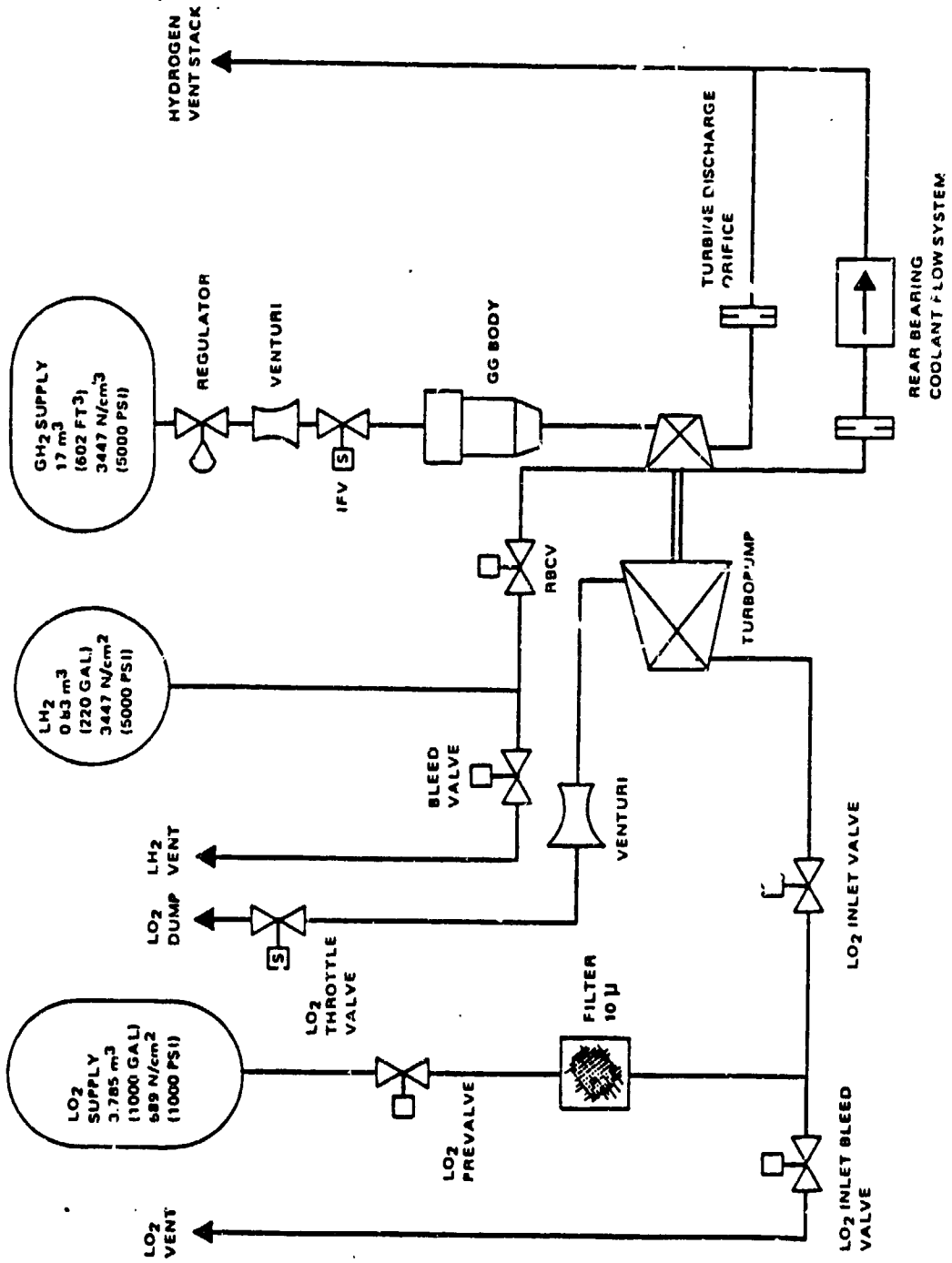


Figure 27. Lima Stand Schematic for MK 48-0 Testing (GH₂ Drive)

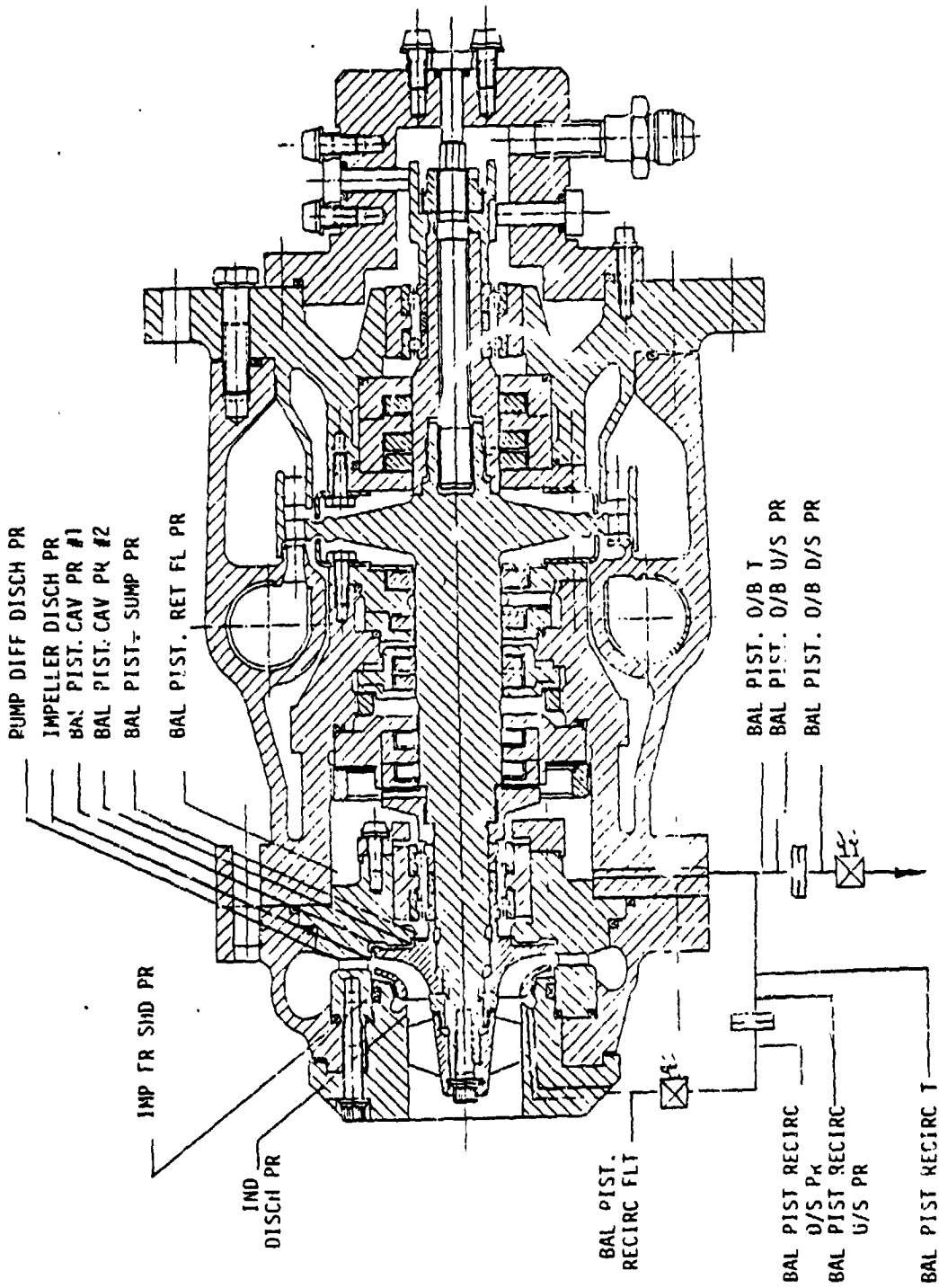


Figure 28. Balance Piston Recirculation and O/B Flow Schematic

Leakage checks were performed on the external joints of the pump and turbine systems. A blowdown was made of the main pump flow loop to verify system integrity and to characterize flow resistance. Instrumentation was calibrated, installed in the facility, and connected to the turbopump ports. A list of measured parameters is presented in Table 11.

The turbopump installation in Lima test cell is shown in Fig. 29 through 31.

Testing. A summary of the tests performed during this test series is presented in Table 12. A total of five tests were made accumulating 161 seconds of operation on the turbopump.

Test No. 001 was aborted prematurely after reaching a speed level of 1487 rad/s (14,200 rpm), due to failure of an accelerometer. Test 002 was conducted at a speed level of approximately 3037 rad/s (29,000 rpm), where satisfactory head flow sweep was realized. Test 003 was initiated by a brief period of operation near 2932 rad/s (28,000 rpm), after which the speed was ramped up to 7226 rad/s (69,000 rpm). At that point the test was terminated by an automatic redline monitoring device because the pump discharge pressure had exceeded the 3447 N/cm² (5000 psig) level. Post-test review of the data showed that the pump operating point was at lower than nominal design Q/N. This, in conjunction with the improved hydrodynamic performance obtained with the modified impeller, caused the discharge pressure to exceed the redline value.

Test 004 was also initiated by a brief dwell at approximately 3142 rad/s (30,000 rpm), after which turbine power was increased to raise the speed to 3037 rad/s (69,000 rpm). All parameters behaved normally during start. Four seconds after reaching the higher speed level an increase in the balance piston pressure readings was noted, with no corresponding change in the pump pressures. Subsequently, the balance piston parameters tracked the pump pressures well. The shift in balance piston pressures was attributed to a possible gas pocket in the system, a temporary bearing hang-up, or some type of blockage in the overboard drain system. The facility lines were visually inspected after Test 004, and the pump exit ports were flow checked with gaseous nitrogen to determine if the overboard drain was blocked. Results were negative.

Analysis of the balance piston position, using the measured values after the shift on Test 004, showed that the piston controlled the axial thrust. The overboard flow was lower but adequate, both for thrust control and bearing cooling. There was no significant rise in the overboard flow temperature which would have been indicative of excessive bearing loads or distress. Neither the Bently proximity indicators nor the casing accelerometer data showed any evidence of adverse behavior. In light of this, testing was continued with Test 005 to define suction capability and characterize performance in a recirculation mode.

TABLE 11. GASEOUS HYDROGEN TURBINE DRIVE

System	Parameter	ID	PID	Range	Headline	Position	DIGR	OSC	FM Tape	Location	Comments
GN ₂	Reg U/S Pressure	PGHR	062	5000 psig		X	X			Fac. Line	
	Venturi U/S Pressure	PGHV	071	5000 psig		X	X				Venturi S/N 9731 P/N VPO31200-SGR
	Venturi U/S Temperature	TGHV	047	0 to 200 F		X					Thermocouple
	Venturi ΔP	PGH*	061	500 psid		X	X				
	GN ₂ Spin Valve U/S Pressure	PSV1	082	5000 psig		X					
	GN ₂ Spin Valve D/S Pressure	PSV2	074	5000 psig		X					
	GN ₂ Spin Valve Position	GHSV	055	Trace		X		X			Meter display for DIGR requirement
Turbopump LOX Outlet Cont.	Throttle Valve Position	TY*	057	Trace		X		X			Meter display for DIGR requirement
	Throttle Valve Outlet Pressure	POTP	100	5000 psig		X					
General	LH ₂ High Pressure Tank Pressure	PHT	087	5000 psig		X	X				
	Facility Duct Pressure	PFX	069	500 psig		X					
	Facility Duct Temperature	TFX	029	0 to 2000 F		X				Fac. Line	Thermocouple
	Axial Proximity Ind	BAP	Tape	Tape FM	X			X	X		
	Radial Proximity Ind No. 1	BRP	Tape	Tape FM	X			X	X		
	Radial Proximity Ind No. 2	BTP	Tape	Tape FM				X	X		90 Degrees from No. 1
	Pump Axial Accelerator	PAA	Tape	Tape FM	X			X	X		
	Pump Radial Accelerator	PRA	Tape	Tape FM	X			X	X		
	Turbo Radial Accelerator	TRA	Tape	Tape FM	X			X	X		
	Pump Speed	RPH	111	100,000 RPM	X	Y	X	X	X		
LOX Pump	Helium Supply Pressure	PHUS	104	0 to 500 psig		X					
	Low Pressure Tank Pressure	POXT	068	500 psig		X	X			Fac. Line	
	Inlet Pressure	POIN	098	200 psig	X	X	X	X			Piezometer Ring
	Inlet Temperature	TOIN	043	-259 to -300 F	X	X	X			Fac. Line	RTB
	Inducer Discharge Pressure	PIND		200C psig		X					
	Impeller Discharge Pressure	PIDP	067	5000 psig		X	X				
	Diffuser Discharge Pressure	PDDP	084	5000 psig		X					
	Pump Discharge Pressure	PDP	089	5000 psig	X	X	X	X		Fac. Line	Piezometer ring X-Y Plotter RTB
	Pump Discharge Temperature	PDT	045	-100 to -300 F		X					
	Balance Piston Cavity Pressure No. 1	P10	096	5000 psig		X	X	X			
	Balance Piston Cavity Pressure No. 2	P11		5000 psig		X					
	Balance Piston Sump Pressure	P12	086	5000 psig		X	X				
	Balance Piston Return Flow Pressure	P13	094	5000 psig		X					
	Balance Piston Return Flow Temperature	PBT	036	-250 to -300 F	X	X	X				Pump Bearing RTB supplied
	D5CH Venturi U/S Pressure	PVUP	08R	5000 psig		X					Venturi S/N 8877 P/N V321059-SGR
	D5CH Venturi U/S Temperature	TVUP	046	-250 to -300 F		X					RTB
	Turbine	D5CH Venturi ΔP	PVDP	063	350 psid		X	X			
Inlet Static Pressure		PT15	065	5000 psig		X	X	X			
Inlet Total Pressure		PT1T	080	5000 psig		X					Kiel Probe Supplied
Inlet Temperature No 1		CT-1	024	0 to 2000 F		X					Wall + 0.150 inch
Inlet Temperature No 2		CT-2	025	0 to 2000 F		X	X				Core Temperature



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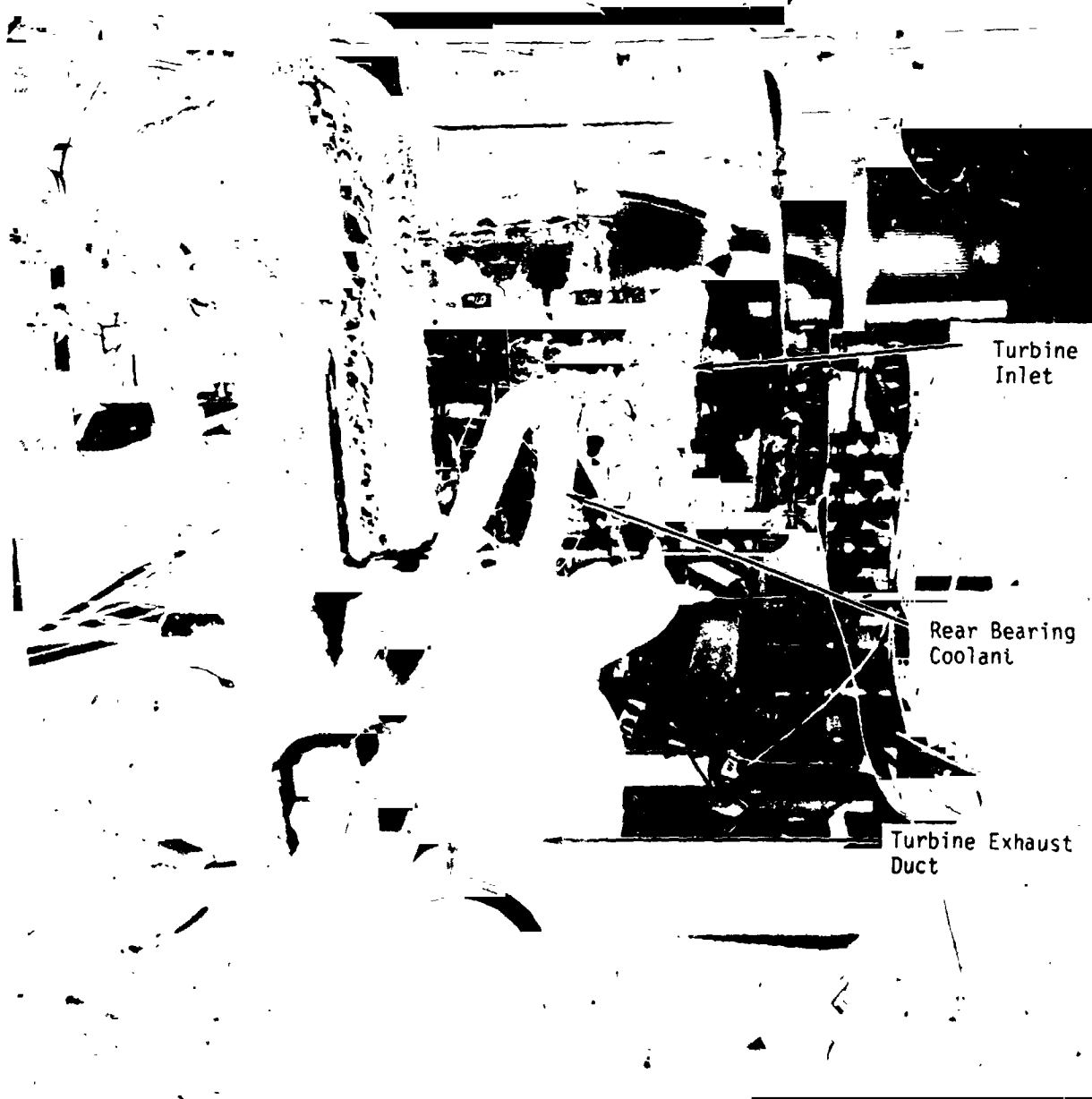
Figure 29. Lima Installation



1HS53-7/20/77-SIC*

Figure 30. Lima Installation

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1HS53-7/22/77-S1*

Figure 31. Lima Installation

TABLE 12. MK 48-0 TEST SERIES NO. 2 SUMMARY

Test No.	Test Date	Duration, seconds	Balance Piston Fluid Routing	Remarks
001	7-21-77	7	Overboard	1487 rad/sec (14,200 rpm) c/o by failed turbine radial acceleration
002	7-21-77	71	Overboard	3068 rad/sec (29,300 rpm) obtained pump H-Q data
003	7-21-77	18	Overboard	2964 rad/sec (28,300 rpm) data. Cut at 7226 rad/sec (69,000 rpm) by pump discharge pressure redline 3447 N/CM ² (5000 psig)
004	7-21-77	33	Overboard	7226 rad/sec (69,000 rpm) H-Q data obtained balance piston pressure shift
005	7-26-77	32	Overboard	7006 rad/s (66,900 rpm) fire damaged pump hardware

Test 005 was initiated with a 20-second duration dwell at 2827 rad/s (27,000 rpm), followed by a ramp to 7006 rad/s (66,900 rpm). The significant pump pressures for Test 005 are shown in Fig. 32. After a brief stabilization at the higher speed level, the pump discharge throttle valve was opened to obtain a high Q/N head-flow data point, before the cavitation test was attempted. At this point of the test, fire was observed in the vicinity of the turbopump, and the test was terminated. The pump end of the turbopump was found to have incurred substantial fire damage, and as a result the unit was removed from the test cell for disassembly and analysis. Damage to the facility was limited to the pump inlet line, instrumentation, and minor tubing.

Incident Investigation

Hardware Analysis. The extent of fire damage incurred inside the turbopump is illustrated in Fig. 33. Shaded areas in the upper half of the section designate those areas where material was consumed by fire. Damage was concentrated in the area of pump inlet, inducer, impeller, diffuser, and pump end bearings. Figure 34 shows the condition of the pump prior to disassembly. The inducer blades were burnt down to the hub, as were the impeller front shroud and vanes (Fig. 35). The outer races, cages, and balls of both pump end bearings were totally consumed. Some burning was also evident in the balance piston return cavity, on the pump side of the slinger, and on the surface of the primary seal housing which is not covered by the slinger.

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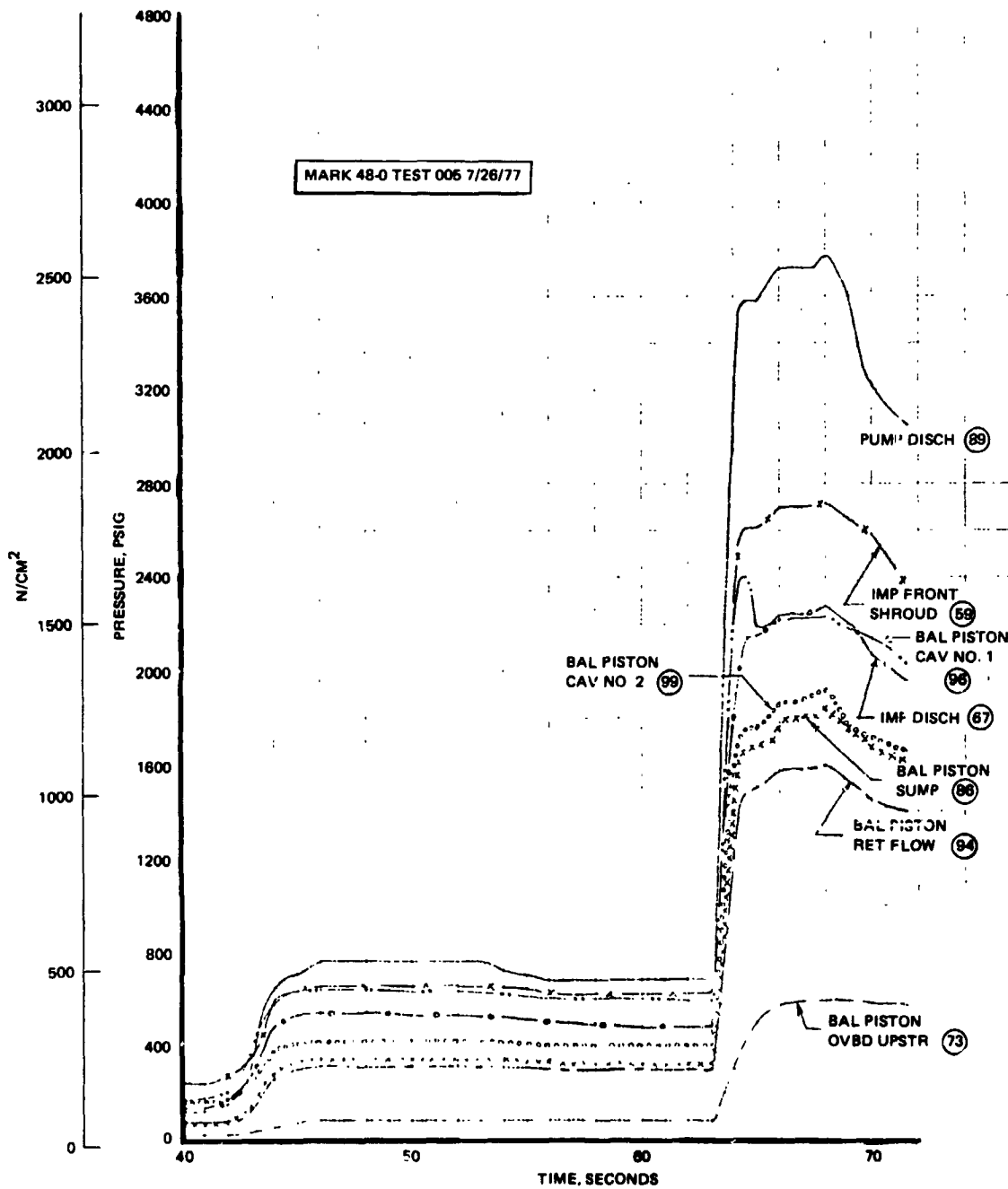


Figure 32. Pump Pressure on Test 005

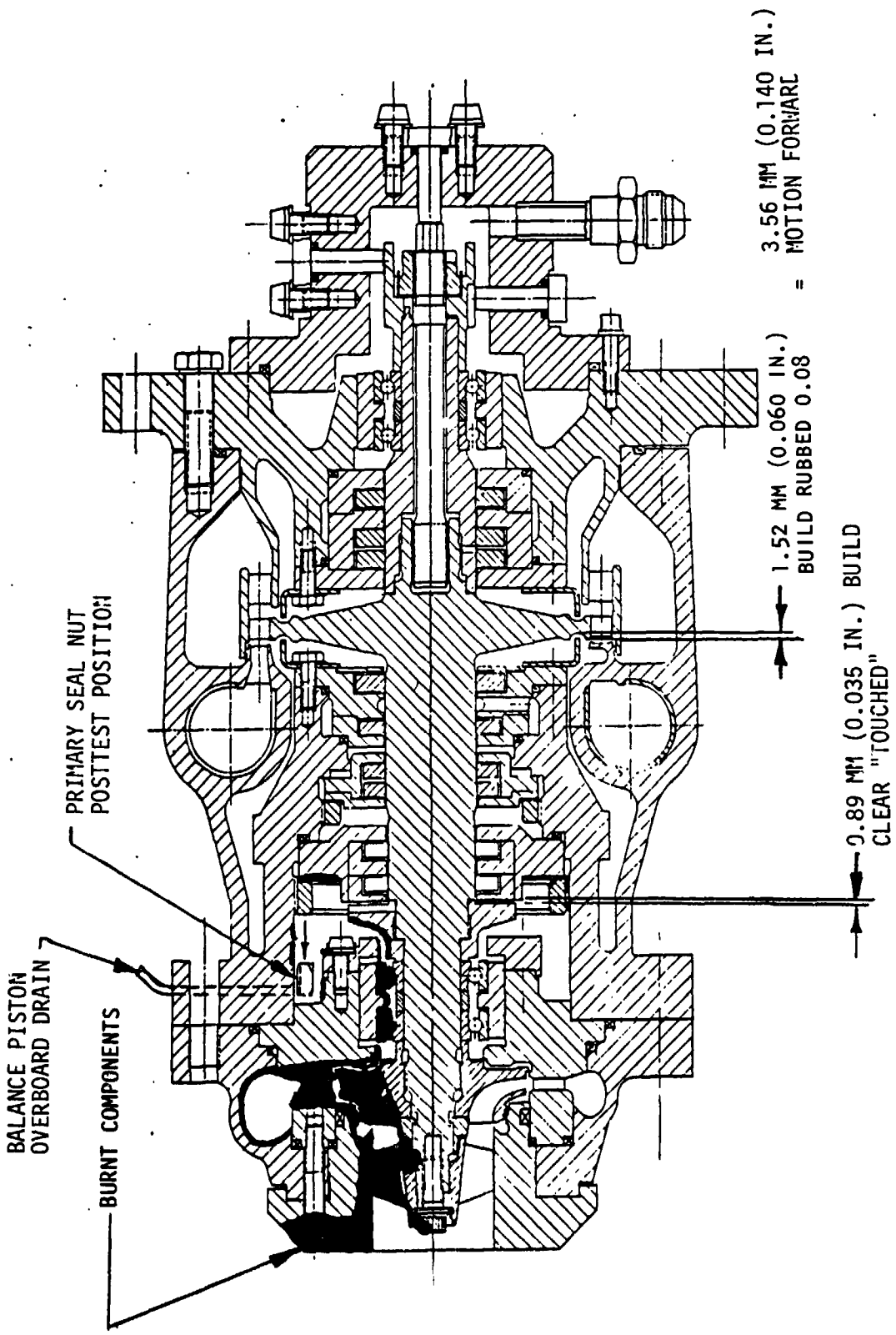
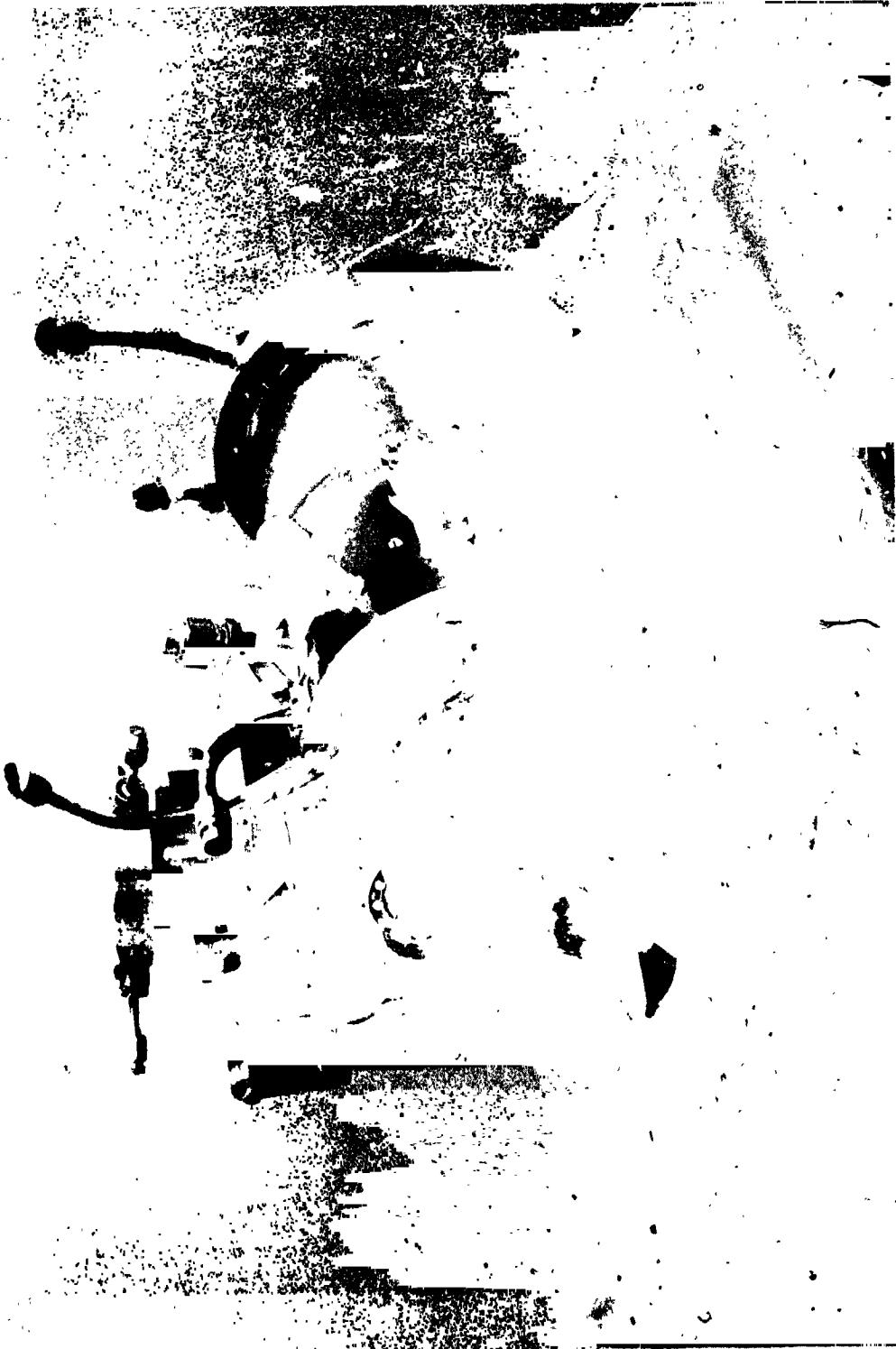


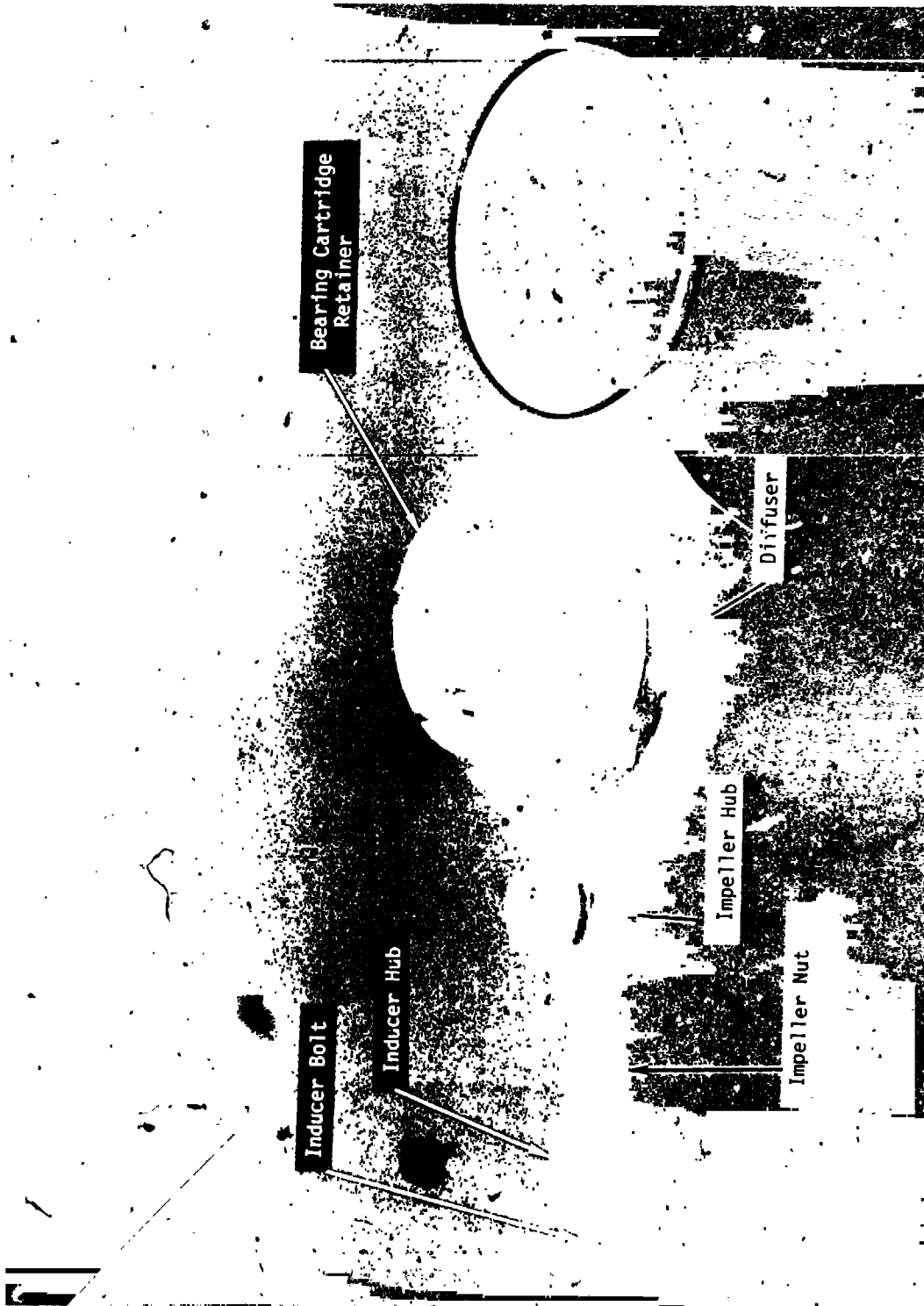
Figure 33. MK 48-0 Turbopump S/N 01-1 Posttest Condition

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1XY55-7/29/77-CLA*

Figure 34. Pump Post Test Condition



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Figure 35. Pump Components

A most significant clue for the cause of the failure was found during disassembly when it was discovered that the primary seal retaining nut had backed off from its thread (Fig. 36) and was located in a position where it severely restricted the overboard flow of the balance piston fluid. The sheet metal stainless-steel lock ring was not found in the cavity, and it is presumed to have been consumed by fire. Build records indicate that it was installed and the locking feature was engaged.

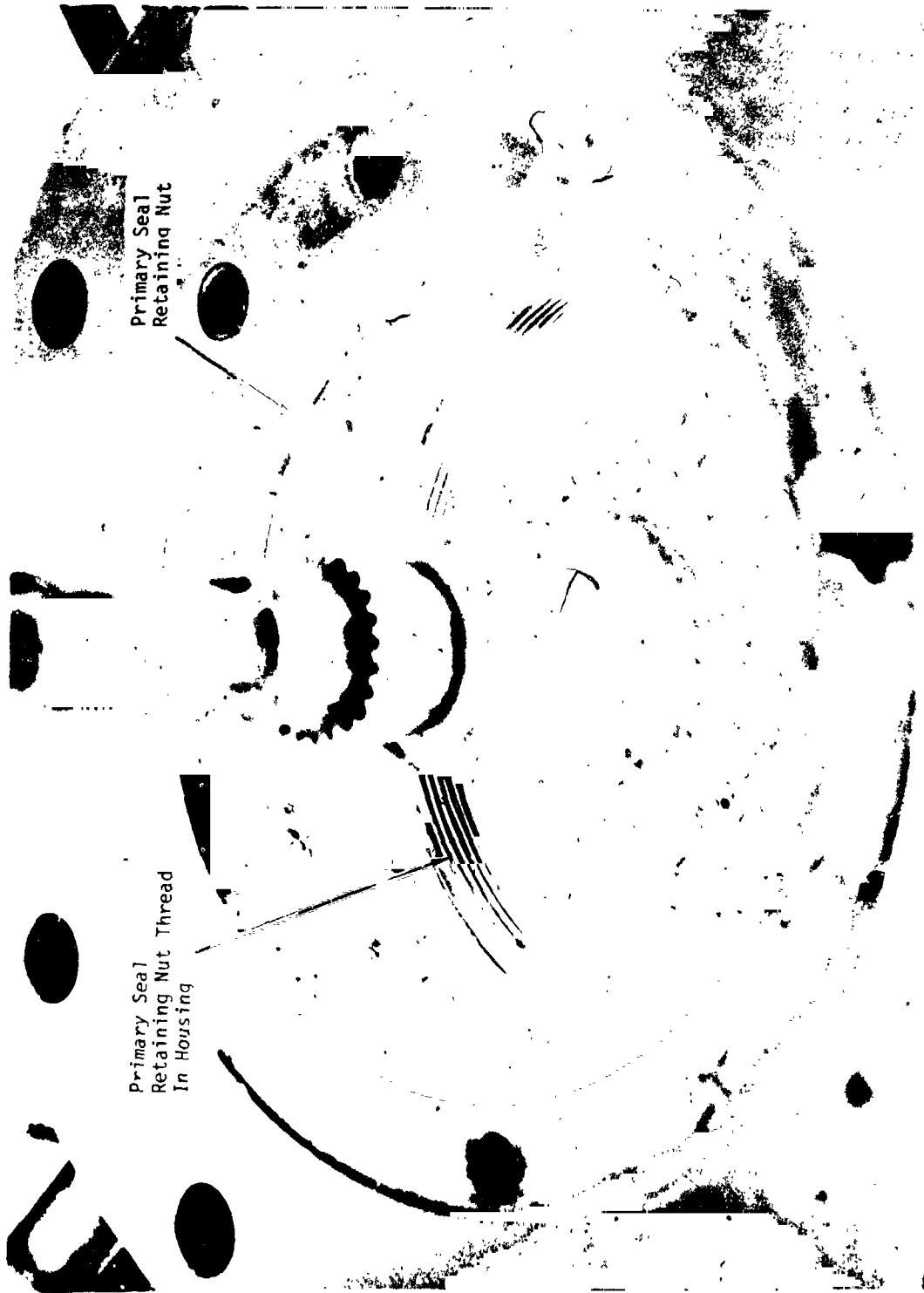
Although the primary seal was not retained over some period of operation, the fluid pressure in the cavity forced it against the housing toward the turbine side, and only superficial contact was between the slinger and the seal housing. The turbine side of the slinger is shown in Fig. 37. Except for some burning on the outer flange, the primary seal working elements were in good condition. As illustrated in Fig. 38, the other shaft dynamic seals were unaffected for the failure.

Minor burn damage was incurred by the rotor (Fig. 39), and it rubbed axially into the pump end hot-gas shield ring and its retaining bolts (Fig. 40). The housing suffered only localized burning around instrumentation ports, which can be repaired. The aft bearing support and turbine end bearing package were unaffected by the failure (Fig. 41 and 42).

Data Analysis. An analysis of the balance piston parameters has been made for various data slices on Tests 004 and 005. The calculated piston relative position as a function of run time is illustrated in Fig. 43. The term χ/δ , plotted along the ordinate, indicates the fraction of the total gap which is present at the high-pressure orifice. Thus at "0", the high-pressure orifice is closed, whereas at 1.0 it is fully open and the low-pressure orifice is closed. The implication from Fig. 43 is that on Test 004 and on the first part of Test 005 the piston position was satisfactory, even after the shift in the piston pressure parameters occurred. However, at the end of Test 005, more than likely as a result of lower impeller discharge pressures due to higher Q/N operation, the piston proceeded to move forward toward the high-pressure orifice, such that at the last data slice before cut the high-pressure orifice clearance was zero, indicating zero control margin.

In a similar fashion, the flowrate through the balance piston was calculated and plotted as a function of run time. As shown in Fig. 44 the calculated flow takes a sharp drop at the end of Test 005, at the same time the high-pressure orifice is closing.

The shift in balance piston pressures incurred on Test 005 has already been discussed above. Thus the data indicate that the primary seal retaining nut actually came loose on Test 004 and caused a partial blockage of the balance piston overboard bleed passages. As a result, the sump pressure increased, causing a reduction in the axial thrust control capability of the piston. Operation at low discharge pressure (high pump flow) on Test 005 further reduced the margin until the piston ran out of balancing range.



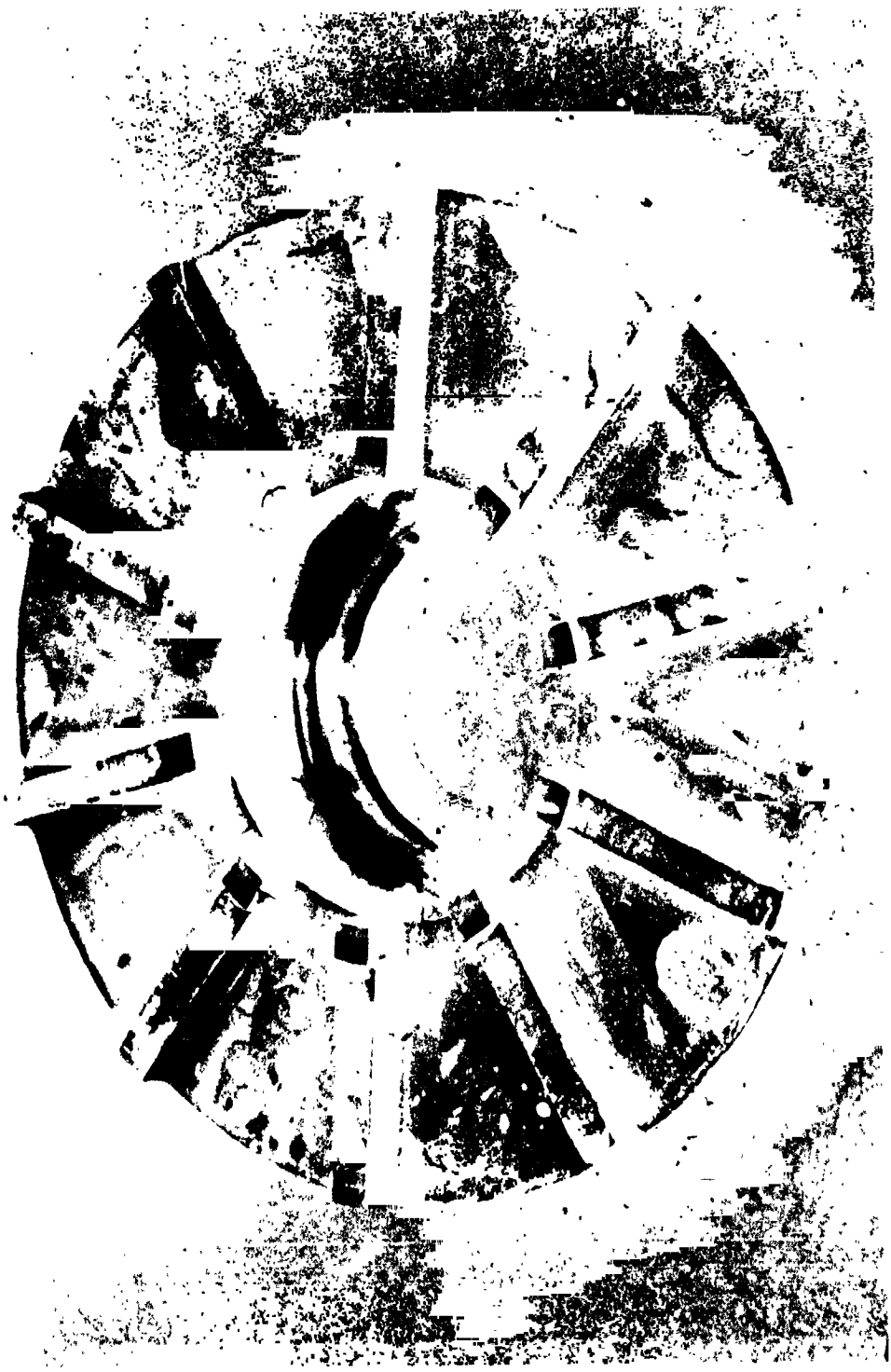
Primary Seal
Retaining Nut

Primary Seal
Retaining Nut Thread
In Housing

IXY55-7/29/77-CIK*

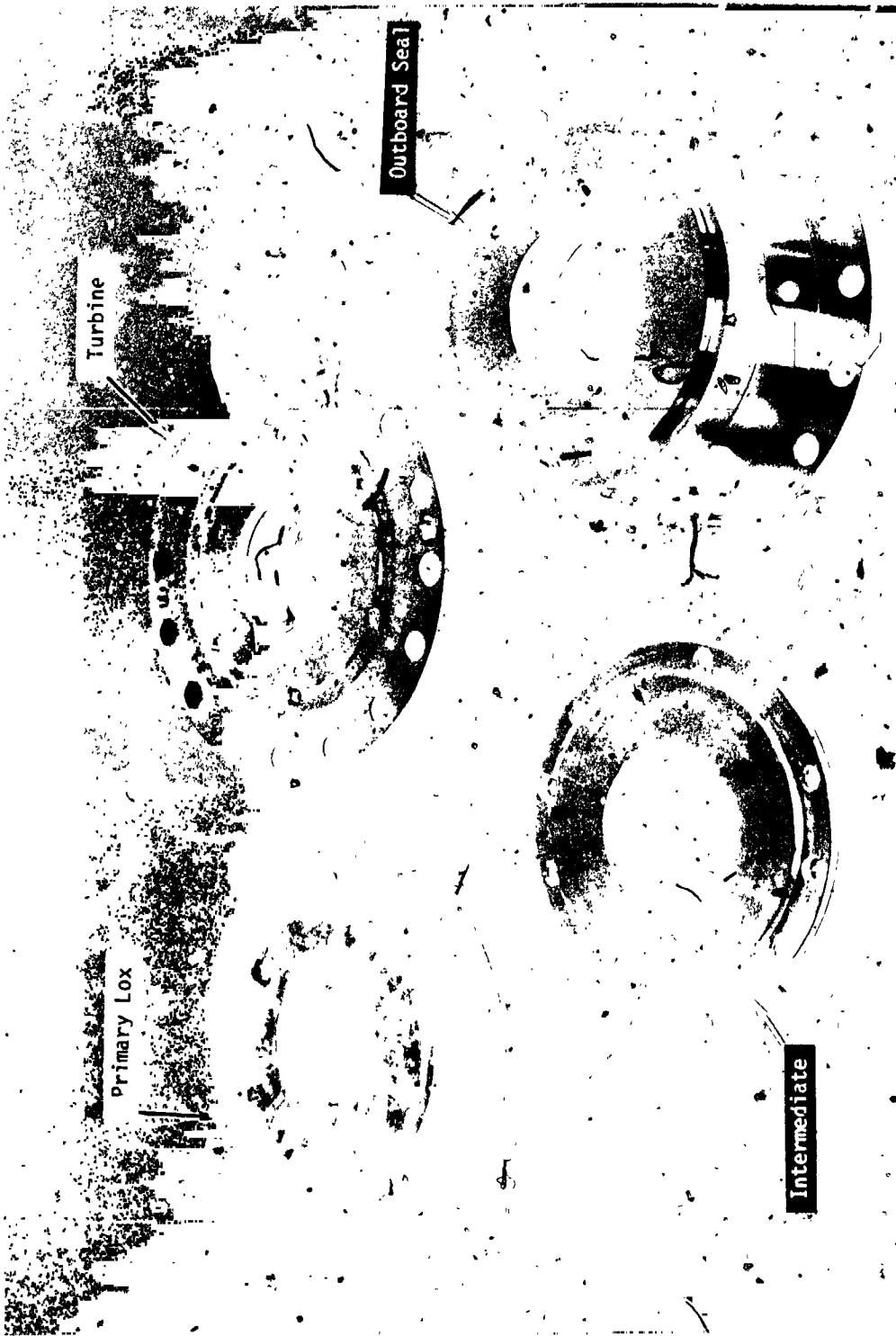
Figure 36. Position of Primary Seal Retaining Nut at Disassembly

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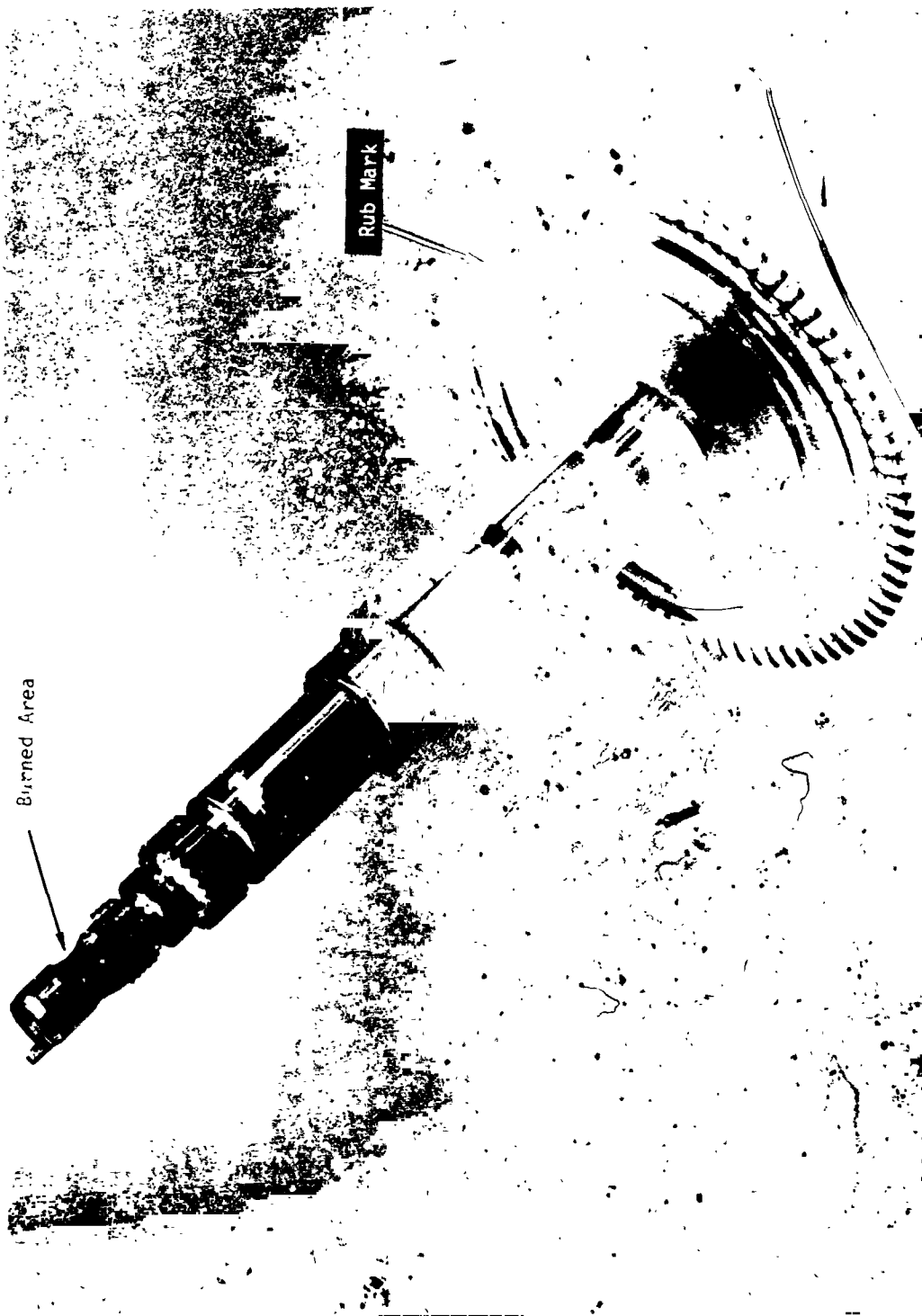
Figure 37. Turbine Side of Slinger



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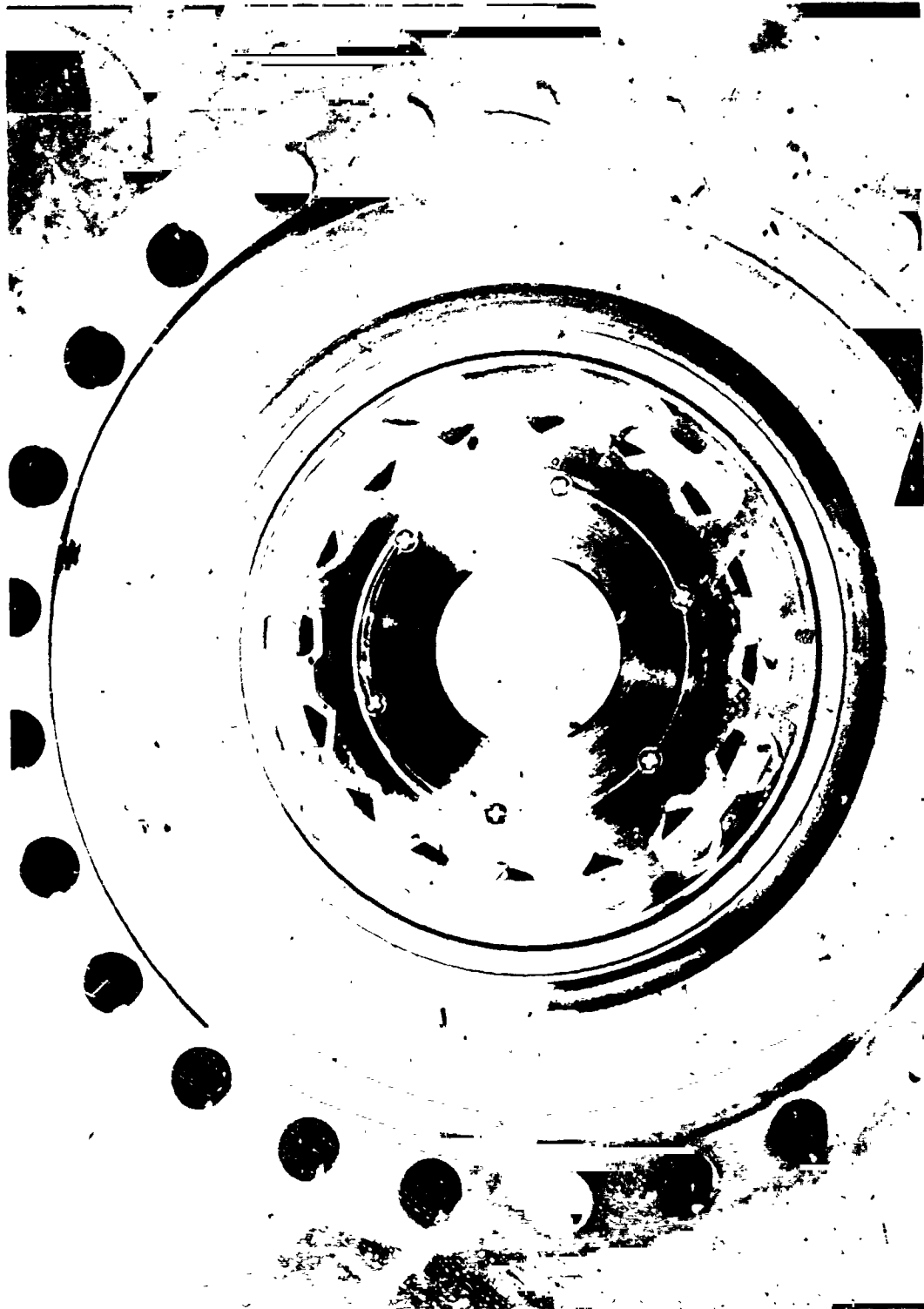
Figure 38. Shaft Seals

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Figure 39. Rotor



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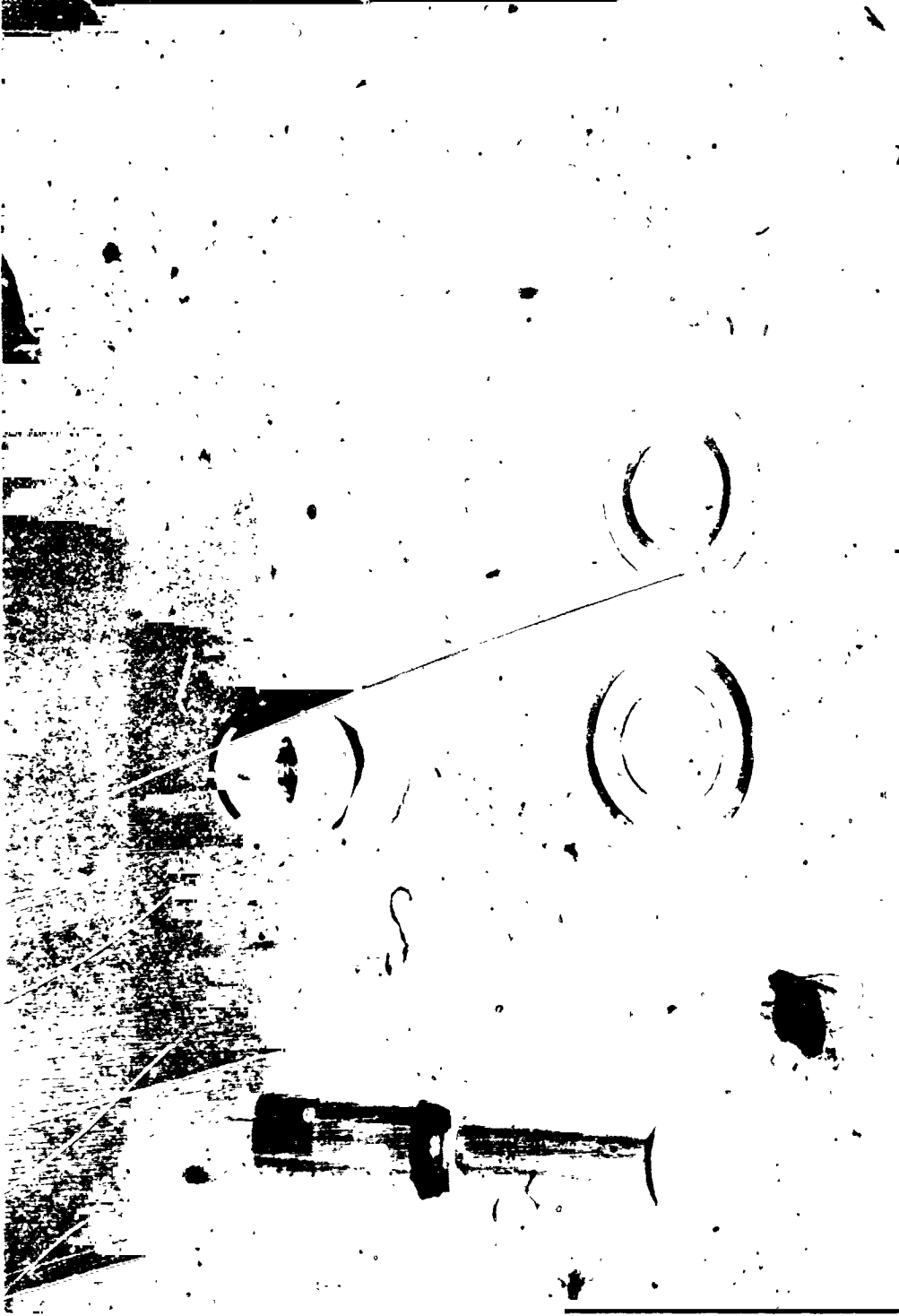
Figure 40. Turbine Nozzle

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Figure 41. Rear Bearing Housing



1XY55-7/29/77-CID*

Figure 42. Aft Shaft and Bearing Package

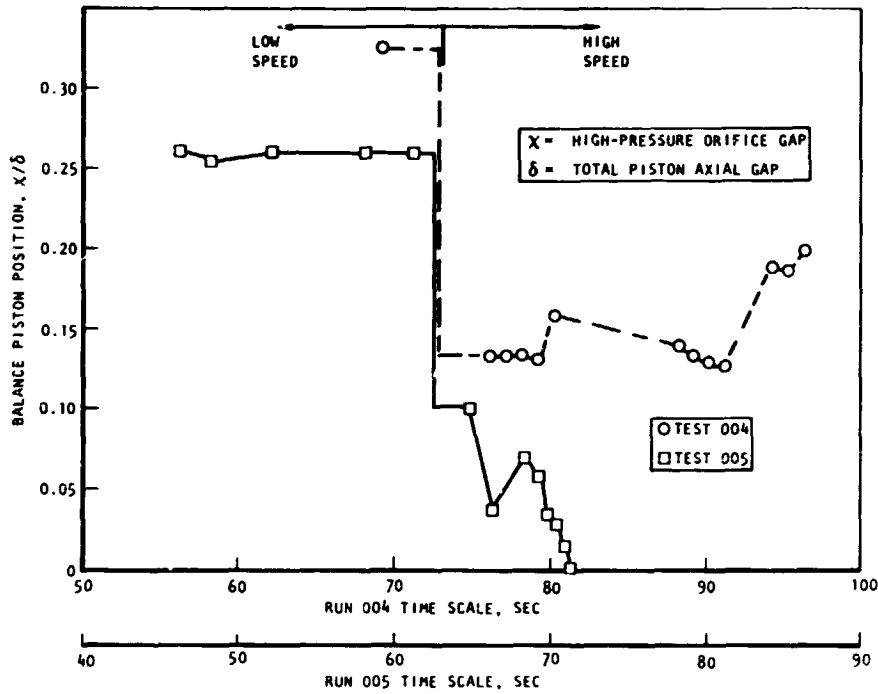


Figure 43. Mark 48-0 Oxidizer Balance Piston Performance Balance Piston Position as a Function of Time

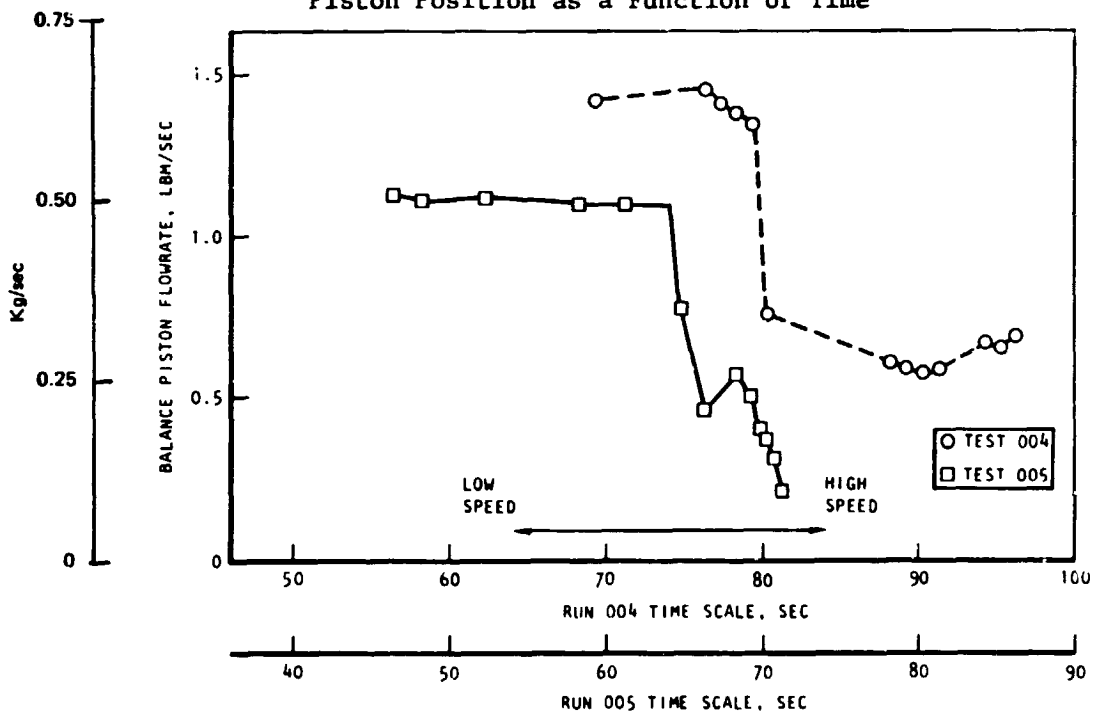


Figure 44. Mark 48-0 Oxidizer Balance Piston Performance Balance Piston Flowrate as a Function of Time

Analysis of the thrust control range prior to blockage of the drain passages indicates adequate margin. Provided the recirculation line losses are maintained at a reasonably low level, recirculating the piston fluid to the inducer discharge (700 psi) should not present a problem to thrust balance. Data analysis revealed no need for a change in the basic balance piston design.

Conclusions and Corrective Action. Based on hardware condition and data analysis, it was concluded tht the moce of failure was the primary LOX seal retaining nut backing off from its thread and blocking the overboard passages for the balance piston fluid. As a result, the axial thrust control capability of the balance piston was greatly diminished and concurrently the coolant flowrate to the pump end bearings was reduced. It is postulated that the pump end bearings were axially overloaded, and as a result they overheated and eventually caught fire. In the process, axial and radial retention of the pump end bearings were lost, resulting in inducer and impeller rubbing and fire.

To prevent a similar occurrence, the direction of the thread in new housings and on the primary seal nut should be reversed such that the fluid vortex from the slinger will tend to tighten the nut rather than back it off. On the existing housing, the thread direction could not be reversed; to permit its continued use in the program, improvements were directed at modifying the nut to reduce the effect of the fluid vortex, and the lock to improve its effectiveness. The details of the modification are described in the next section of this report.

CORRECTIVE DESIGN CHANGES

The second test series conducted on the Mark 48-0 turbopump revealed a problem with the primary seal retaining nut (Fig. 45), in that the fluid exiting from the slinger (with a substantial amount of whirl) directly impinges on the torque slots of the nut, and in the process imposes a large loosening torque on the nut. In its original configuration, the nut acted as an antivortex ring. This actually resulted in the nut backing off on the last test series, and caused substantial hardware damage.

To prevent the possibility of a similar occurrence, a new nut, shown in Fig. 46, was designed on which the torque slots are shielded from the fluid whirl on the inner diameter by continuous material. Although more expensive to fabricate, this should greatly reduce the amount of torque imposed on the nut by the fluid. In addition, the lock ring thickness was increased from 0.41 mm (0.016 inch) to 0.81 mm (0.032 inch); the number of locking slots was doubled from two to four; and the lock slot contour in the nut has been changed from an arc to one including orthogonal sides.

The modified configuration of the primary seal nut and lock ring is illustrated in Fig. 47.

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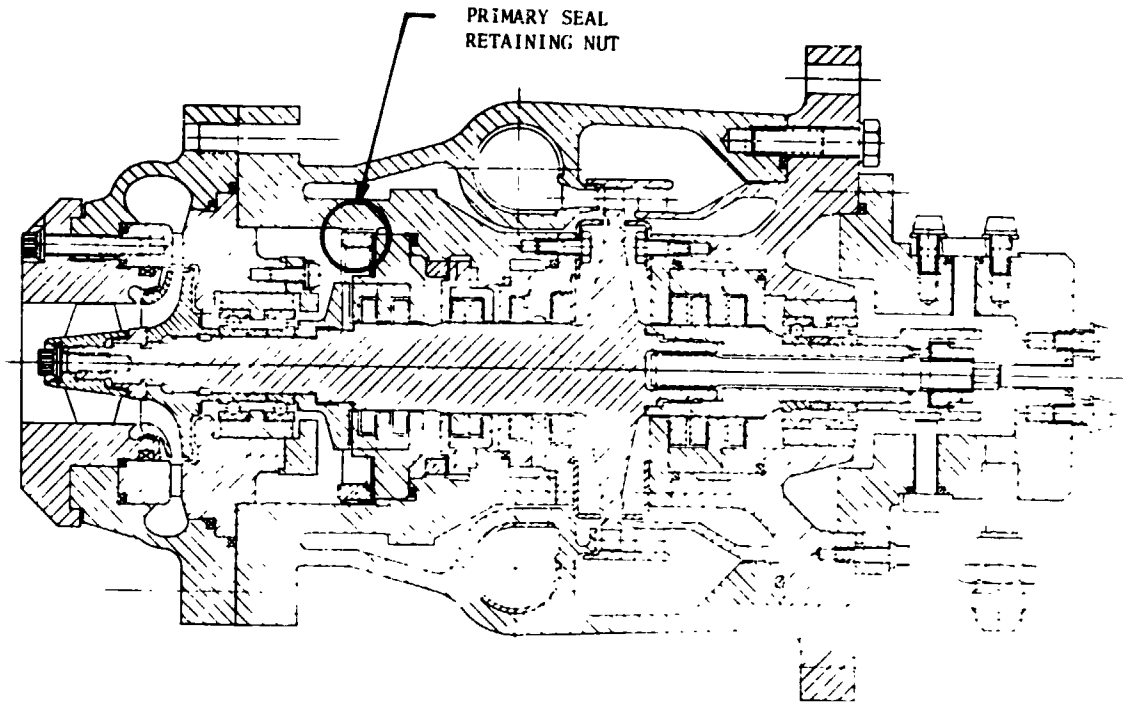


Figure 45. Mark 48-F Turbopump Assembly

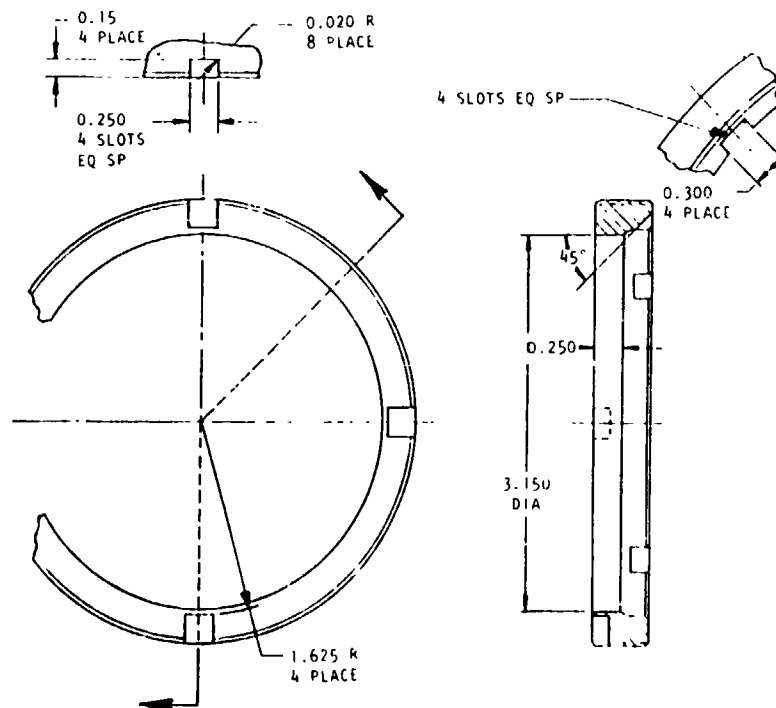
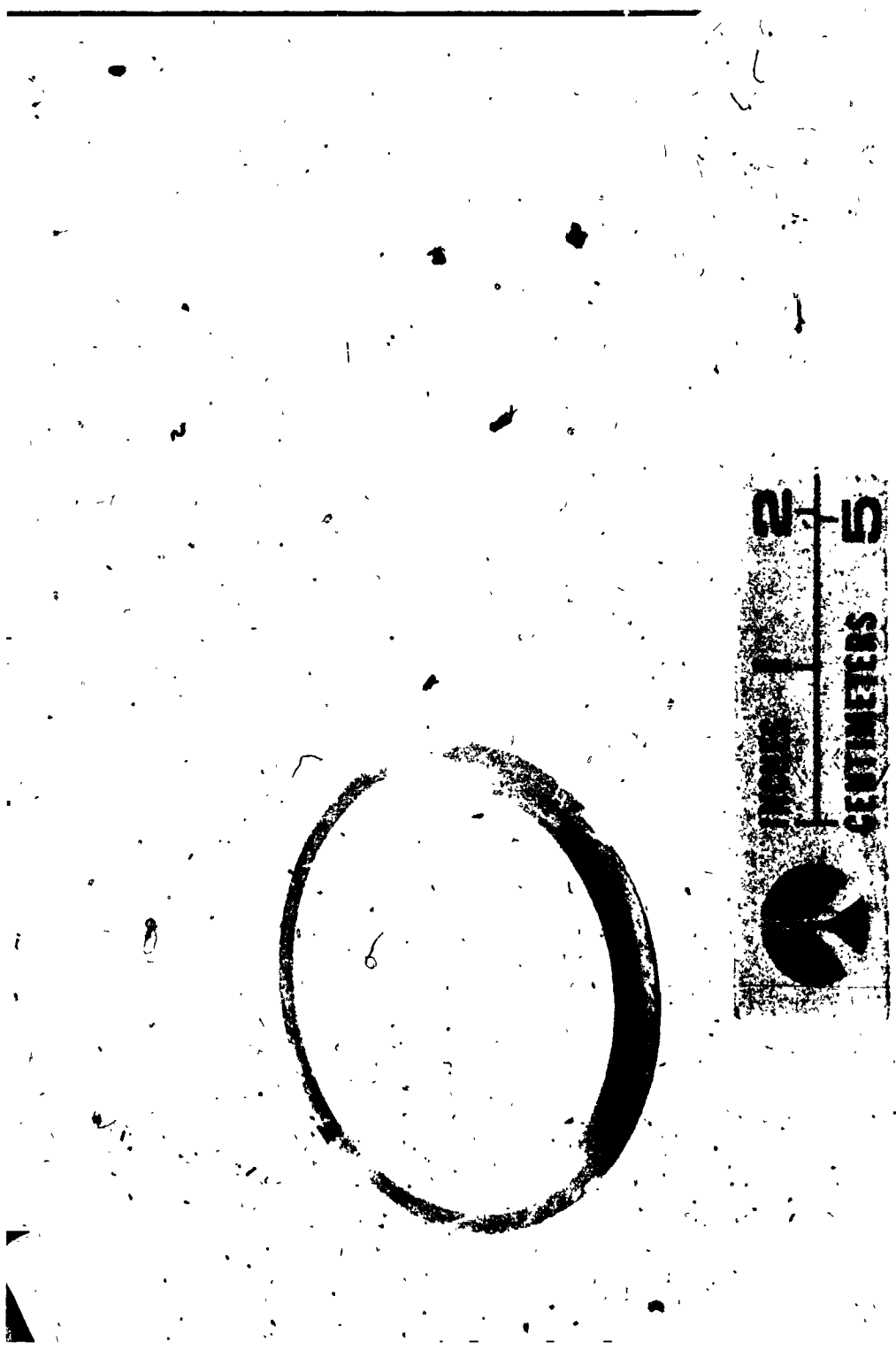


Figure 46. Modified Primary Seal Nut Design



1XY52-10/20/77-CIB*

Figure 47. Mark 48-0 Primary Seal Retaining Nut and Lock Ring
(Open Side)

TURBOPUMP S/N 02-0 ASSEMBLY

During the second test series conducted in July 1977, sufficient information was obtained to show that the modifications which had been incorporated in the impeller produced a noticeable improvement in the pump performance. Nevertheless, the limits of suction capability were not defined, and the effect of returning the balance piston flow to the impeller inlet was not evaluated. It was therefore decided to employ the same hardware configuration in the third test series, with the exception of modifications which are needed to prevent the primary seal nut from backing off.

Thus, the spare set of hardware was modified to include the following features:

1. The impeller inlet area was increased by enlarging the eye diameter and trimming back the blade leading edges.
2. An inducer static pressure port was added.
3. The diffuser-to-inlet housing radial clearance was reduced by plating to eliminate potential internal leak path.
4. An overboard bleed port was incorporated in the housing to facilitate routing the balance piston fluid externally.
5. The balance piston internal recirculation passages were blocked with pins in the inlet housing.
6. Passages and manifolding were included in the inlet housing to provide the capability of returning the balance piston fluid to the impeller inlet from an external source.
7. A new primary seal retaining nut was fabricated in which the torque slots are shielded from the swirling fluid.
8. A more substantial lock was fabricated for the primary seal, with improved locking features.
9. The slinger was modified to obtain a 0.735 mm (0.030-inch) axial clearance on the primary seal side, and the seal housing was silver plated to reduce the hazard in the event of an inadvertent contact.

Balancing and assembly of the turbopump was accomplished in accordance with the procedure described in Ref. 1. Runouts of the critical surfaces of the balance assembly were measured, and values noted in Fig. 48. Measurements of critical clearances and fits taken during the buildup of the turbopump are summarized in Fig. 49 through 52.

TEST SERIES NO. 3 (1978)

Test Plan

The objective of the third test series, conducted under this contract, was to perform a checkout test at 3142 rad/s (30,000 rpm), and in the process obtain heat-flow characterization data at that speed. This was to be followed by pump

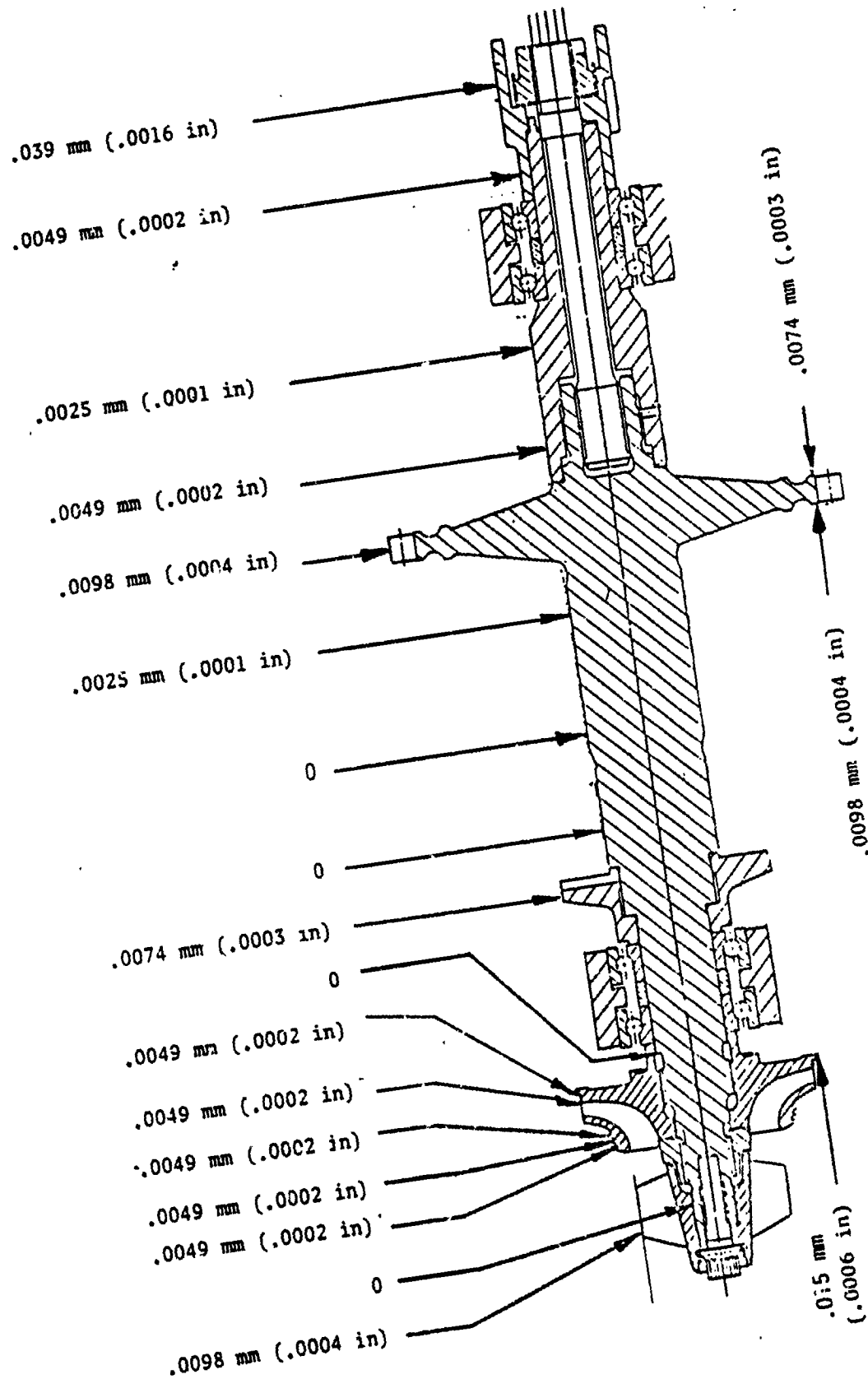


Figure 48. MK 48-0 Turbopump S/N 02-2 Balance Assembly Runouts

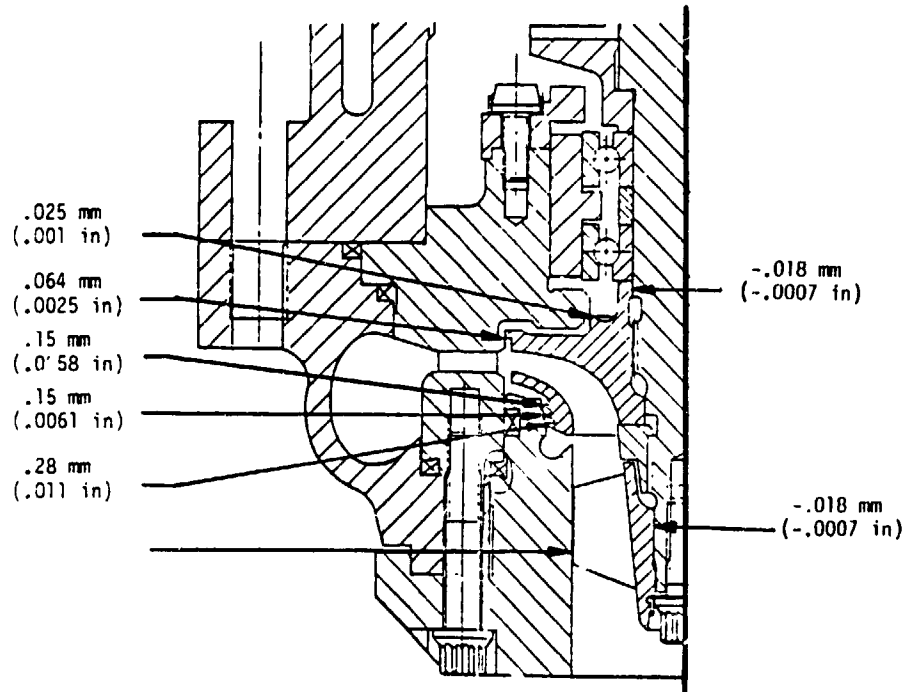


Figure 49. MK 48-0 Turbopump S/N 02-0 Pump
Diametral Clearances and Fits

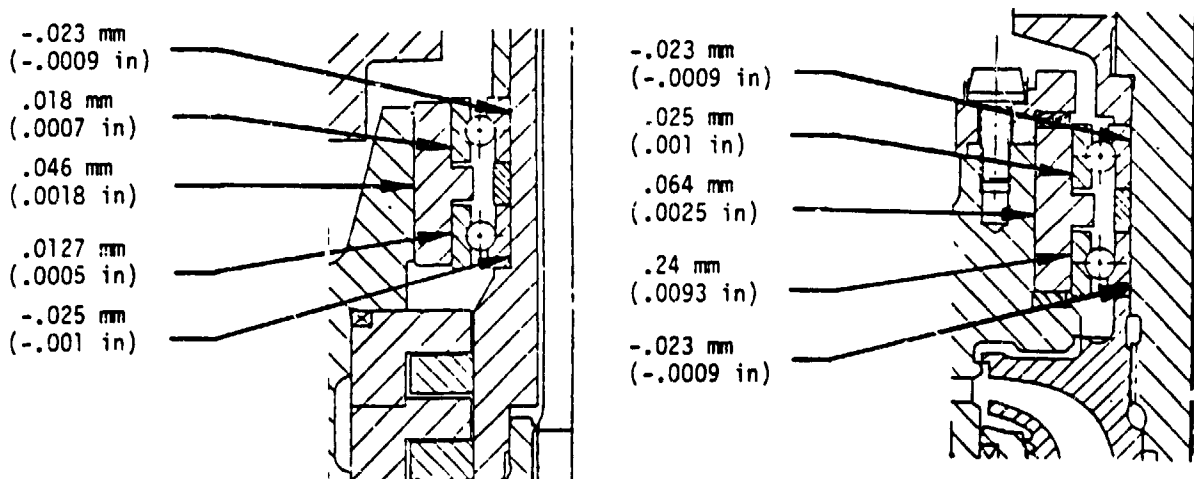


Figure 50. MK 48-0 Turbopump S/N 02-0 Bearing Fits

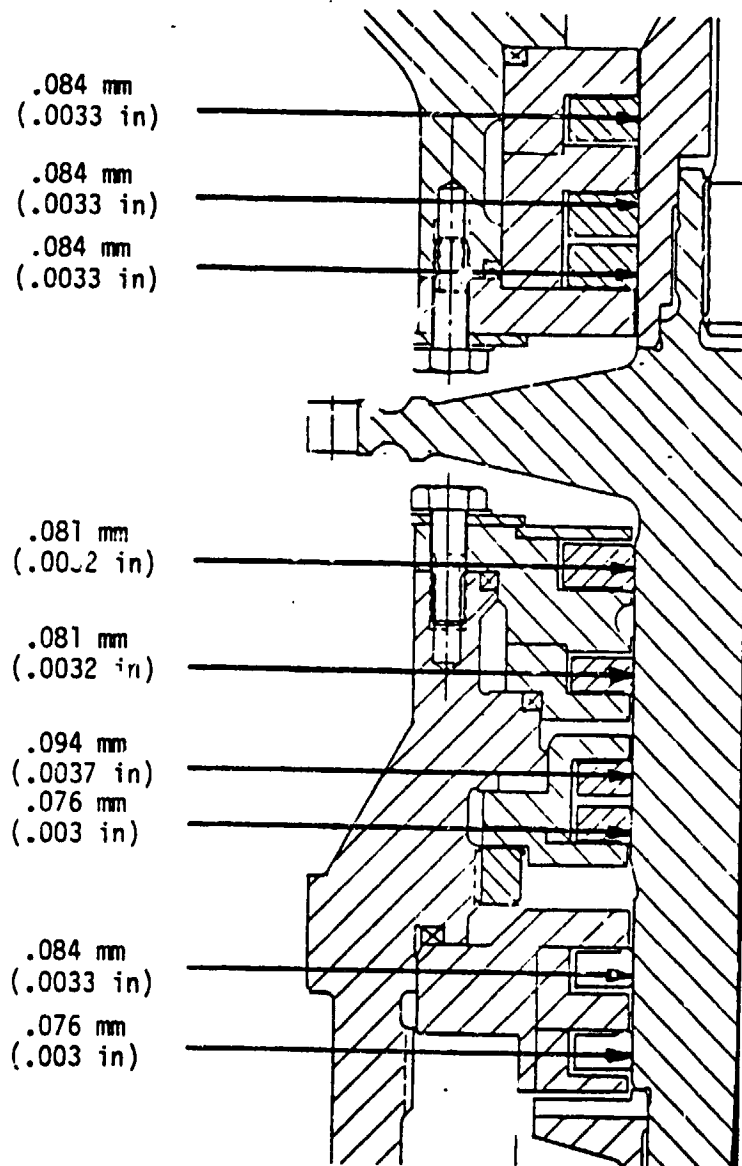


Figure 51. MK 48-0 Turbopump S/N 02-0
Seal Diametral Clearances

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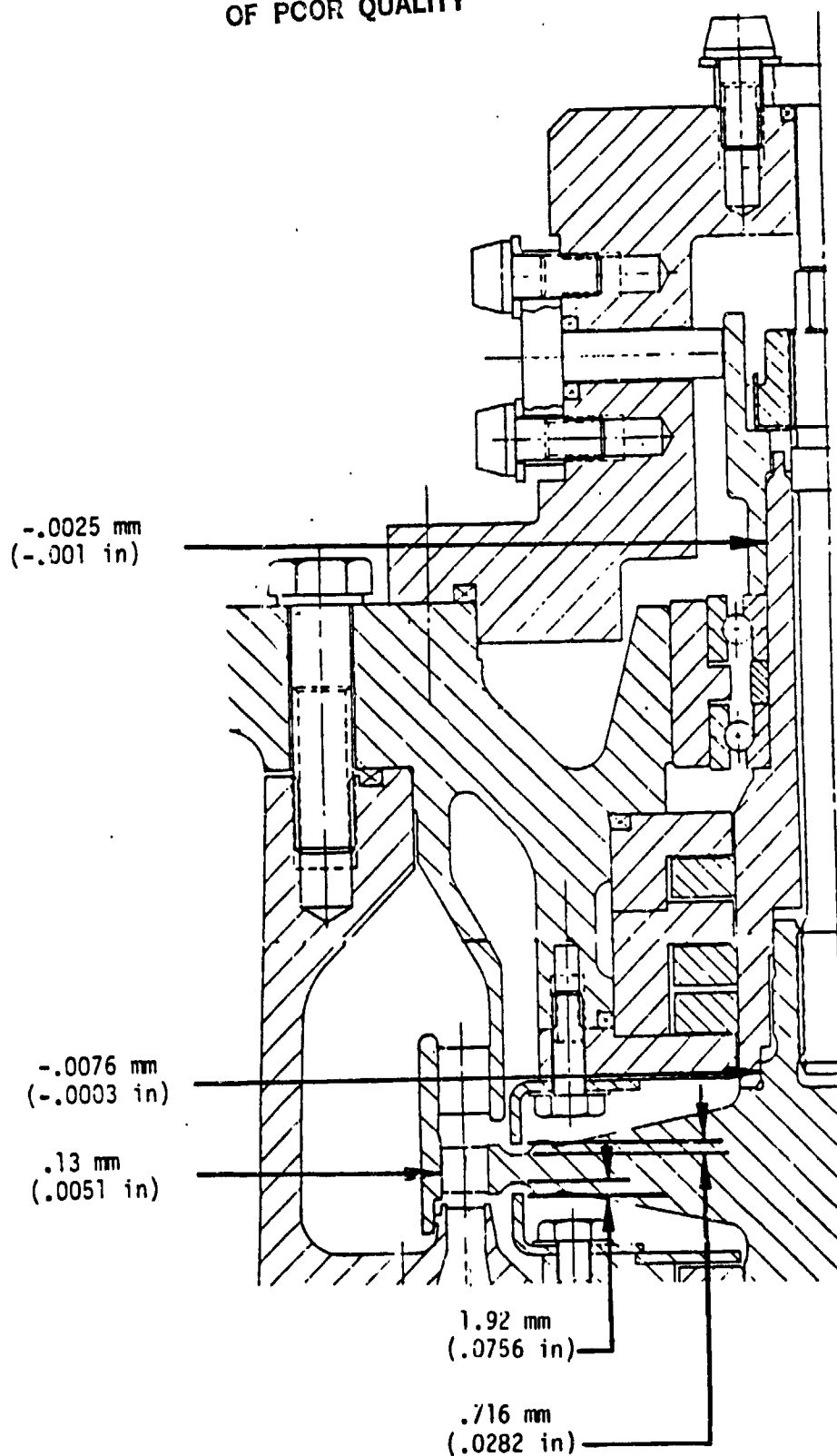


Figure 52. MK 48-0 Turbopump S/N 02-0 Turbine End Clearances and Fits

characterization data at the design speed of 7330 rad/s (70,000 rpm). Subsequently, the test series was to be continued under a new contract, the objective of which was to obtain LOX primary seal performance information.

During these initial two tests, the balance piston fluid was planned to be routed overboard, and the turbine was to be driven by ambient-temperature gaseous hydrogen.

The turbopump instrumentation, as well as the facility installation, was the same as those described previously for Test Series No. 2.

Test Description

Four tests were conducted during this series (Table 13). The first was prematurely terminated by the VSC device at a pump speed of 3089 rad/s (29,500 rpm); total duration was 31 seconds. Vibration data were thoroughly evaluated, and the VSC cut off was found to be erroneous. Saturation of an amplifier was suspected and that amplifier was replaced.

TABLE 13. MK 48-0 TEST SERIES NO. 3 SUMMARY

Test No.	Test Date	Duration, seconds	Balance Piston Fluid Routing	Remarks
003	5-19-78	31	Overboard	Faulty VSC cut at 3089 rad/s (29,500 rpm)
004	5-23-78	124	Overboard	Satisfactory H-Q test at 3142 rad/s (30,000 rpm)
005	5-25-78	38	Overboard	Faulty VSC cut at 5760 rad/s (55,000 rpm)
006	5-31-78	43	Overboard	H-Q data at 6911 rad/s (66,000 rpm), cut by pump bearing coolant redline

The second test conducted for 124 seconds. A speed versus time plot for this test is shown in Fig. 53. Speed was varied by manually controlling the turbine inlet servovalve position. After the turbopump was satisfactorily chilled, speed was increased to 942 rad/s (9000 rpm) for a duration of 10 seconds, after which it was increased to 3142 rad/s (30,000 rpm), making adjustments in pump discharge valve position as required to operate approximately at nominal Q/N. At 3142 rad/s (30,000 rpm), the target value of the automatic feedback control system of the turbine inlet valve was set to the turbine inlet pressure actually displayed on the strip chart, and the feedback control system was activated to maintain constant turbine inlet pressure.

To reduce the recorded parameters to a usable form, a computer program was utilized which received the Beckman data acquisition unit information and converted it to the desired form. To illustrate the output obtained, the printouts for two tests are included in Appendix C: Test No. 004 showing ambient gaseous hydrogen drive data for a satisfactory H-Q test at 3142 rad/s (30,000 rpm), and Test No. 006 showing ambient gaseous hydrogen drive for a satisfactory H-Q test at 6911 rad/s (66,000 rpm).

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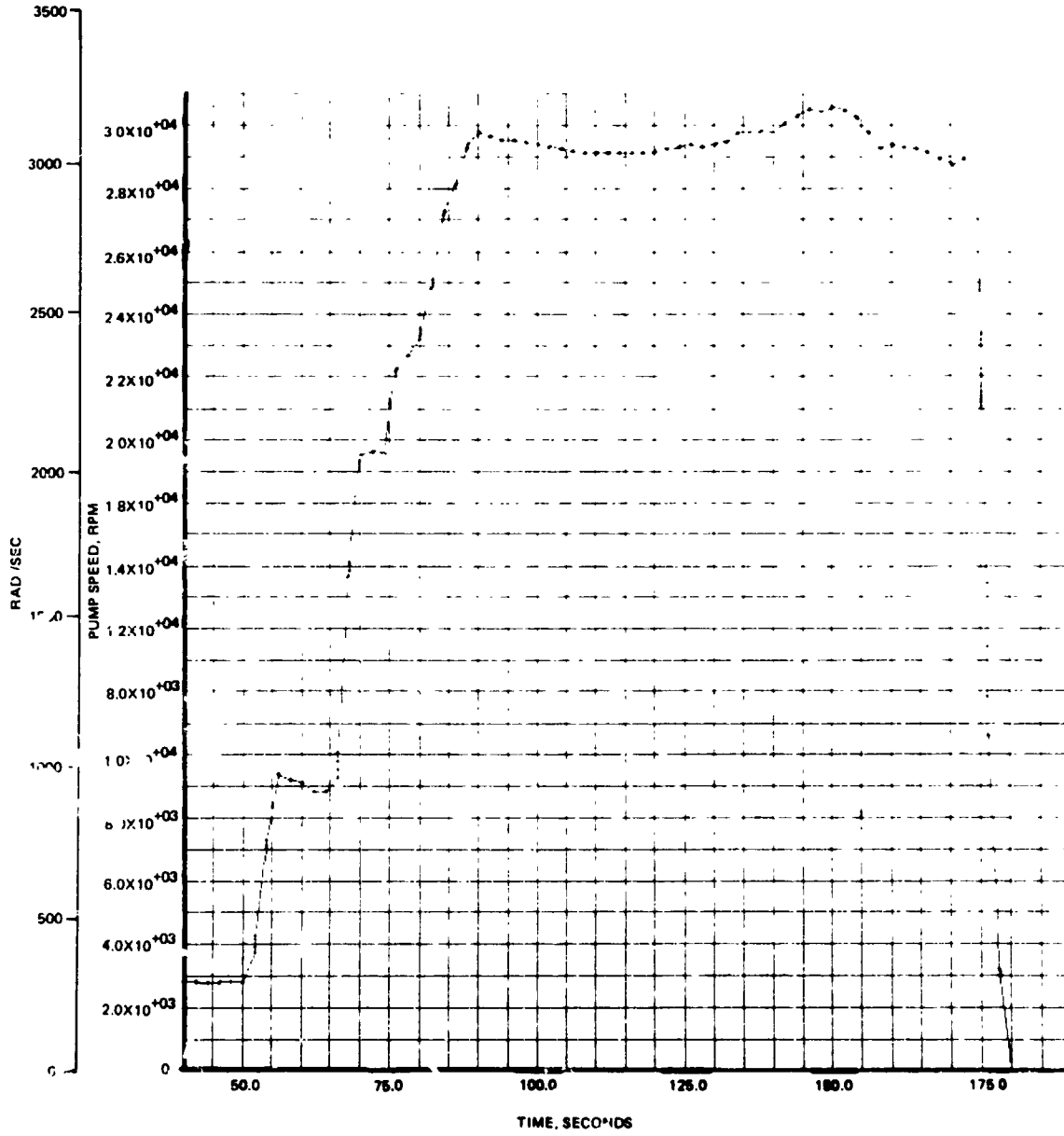


Figure 53. Speed Plot for Test No. 004

The throttle valve located in the facility downstream of the pump discharge was varied to characterize the pump head-flow relationship. The operation encompassed a flow range extending from 71% to 109% of the design flow-to-speed ratio (Q/N).

The third test was conducted for 38 seconds, and was terminated on the speed ramp from 3142 rad/s (30,000 rpm) to 7330 rad/s (70,000 rpm), at approximately 5760 rad/s (55,000 rpm) by the VSC device. Again, the VSC cut-off was found to be erroneous. In going through the second critical speed at approximately 5236 rad/s (50,000 rpm), the small signal increase in accelerometer G level again saturated the VSC device. As a result, the entire VSC system was reworked.

A fourth test was conducted for a total of 43 seconds of which the last 8 seconds were at 6912 rad/s (66,000 rpm). A plot of the rotor speed as a function of time is included in Fig. 54 for the fourth test.

With the pump thoroughly chilled with LOX, rotor speed was slowly increased to 3142 rad/s (30,000 rpm), at which point the automatic turbine inlet pressure control servosystem was activated. After approximately 20 seconds of operation at 3142 rad/s (30,000 rpm), the speed was increased to 6912 rad/s (66,000 rpm). The throttle valve setting in the pump discharge system was varied to obtain head-flow data. After approximately 8 seconds of operation at the increased speed level, the test was terminated because the pump bearing coolant temperature exceeded the 144 K (260 R) maximum redline. The reworked VSC system performed in excellent fashion. The hydrodynamic data obtained was primarily at low-flow conditions; the pump Q/N ranging from 80% to 95% of design value.

After the fourth test, the turbopump rotor exhibited a higher than normal torque; as a result, the turbopump was removed from the test stand to evaluate the higher-than-anticipated bearing temperature and high-rotor torque.

HYDRODYNAMIC PERFORMANCE ANALYSIS

Pump Performance, Test Series No. 2

During the second test series, sufficient steady-state data were obtained both at low speeds and near design speed to define the non-cavitating head-flow characteristics of the pump, and to provide an indication concerning its suction performance capability. Measured head-flow data are plotted in Fig. 55. The data show close agreement with predicted values near the design point, but the actual slope of the curve is slightly steeper than predicted. Within the anticipated flow excursion range of the Advanced Space Engine, the steeper slope does not present a problem. The data group located at the highest flow-rate include both high- and low-speed data, which indicate that the steeper slope at high flowrates is not cavitation-related.

The isentropic efficiency values, calculated on the basis of fluid temperature rise in the pump, are plotted in Fig. 56. A peak efficiency of 68% is achieved slightly below the .0156 m³/s (232 gpm) design flow. Calculated efficiency values, based on turbine performance, yielded similar results with slightly more data scatter. The obtained efficiency is considered excellent for a pump in this size range.

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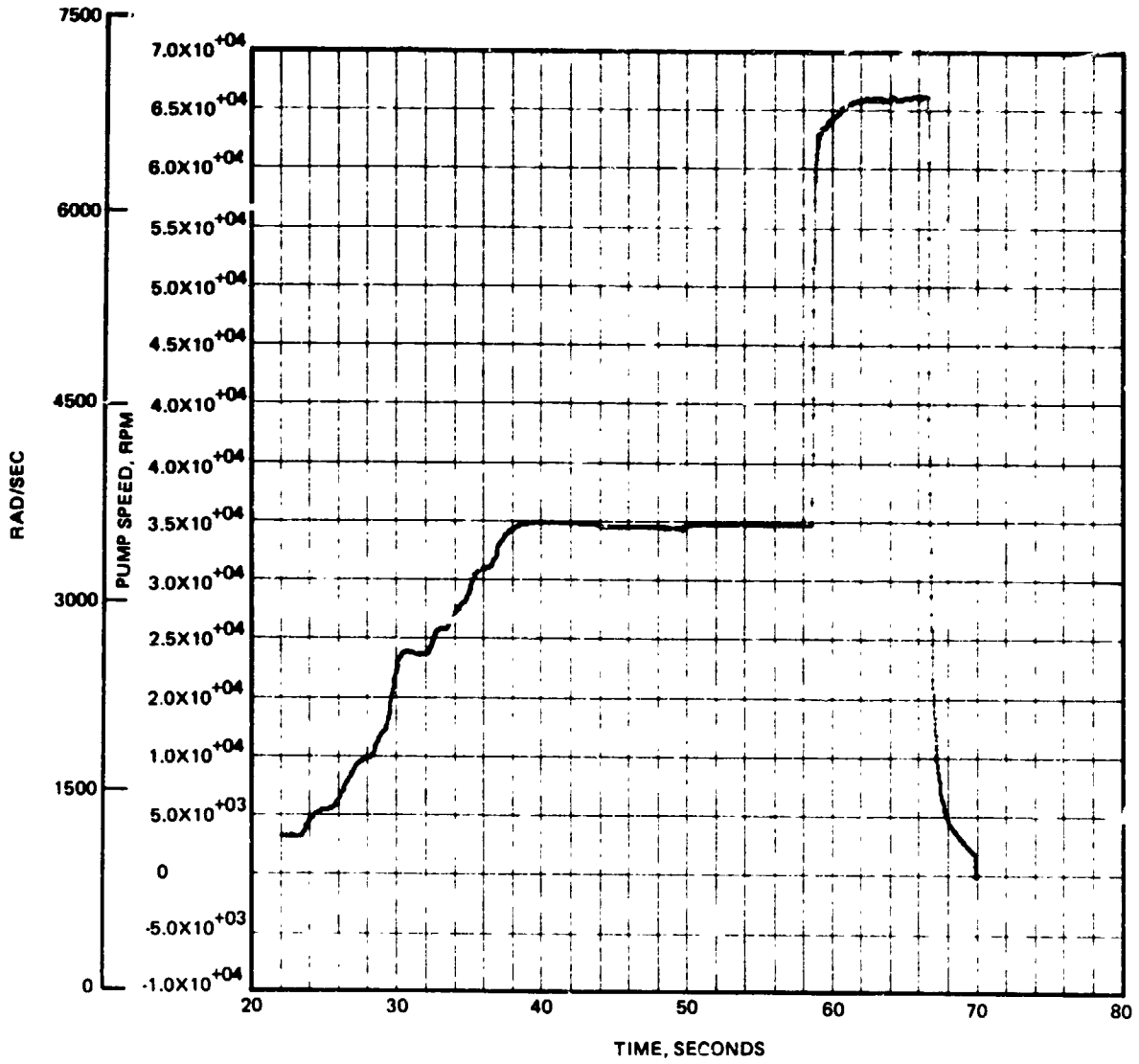


Figure 54. Speed Chart for Test No. 006

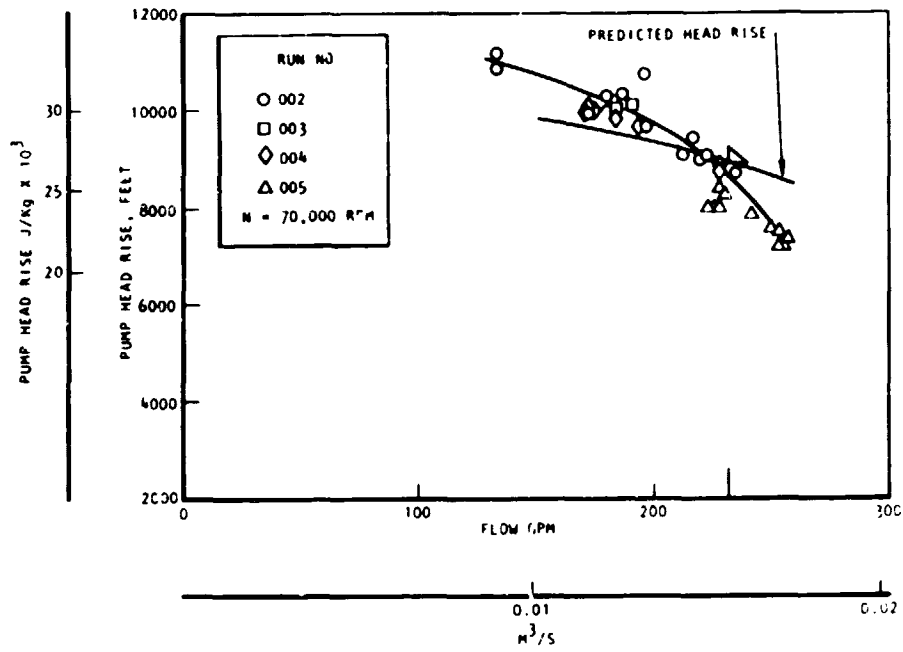


Figure 55. Mark 48 Oxidizer Pump

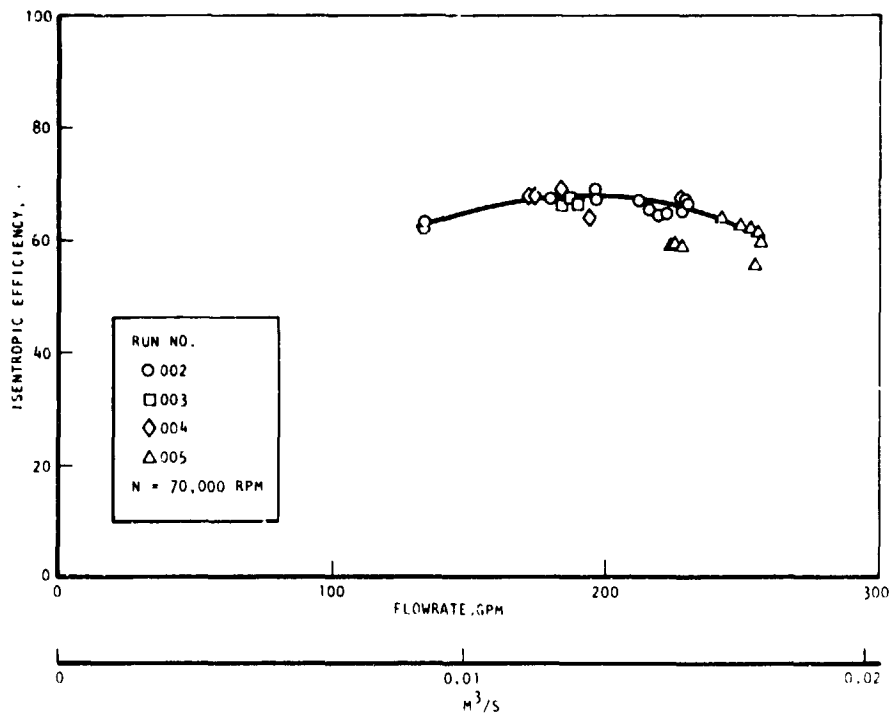


Figure 56. Mark 48-0 Isentropic Efficiency

Limits of cavitation performance were not defined before testing was terminated as a result of the failure; however, operation up to a suction specific speed of $85263 \left[\frac{\text{rad}}{\text{s}} \left(\frac{\text{m}^3}{\text{s}} \right)^{1/2} / \left(\frac{\text{J}}{\text{kg}} \right)^{3/4} \right] \left\{ 24,300 \left[\text{rpm}(\text{gpm})^{1/2} / \text{ft}^{3/4} \right] \right\}$

was realized at a flow coefficient of 0.094 at the end of Test 005 without any evidence of cavitation. Prior to the impeller modification, cavitation with a substantial head loss was encountered at much lower suction specific speed values. Thus unquestionably the impeller modification improved the hydrodynamic performance of the pump to a great degree. Although the NPSH limits of the pump are still to be defined with and without balance piston fluid recirculation, the noncavitating hydrodynamic characteristics of the pump meet the Advanced Space Engine requirements satisfactorily.

Hydrodynamic Analysis, Test Series No. 3 (May 1978)

The MK 48 oxidizer pump has not undergone any design changes in the pumping elements since the second test series of July 1977. There have been no changes to the inducer, impeller, vaned diffuser, or volute. There have been some modifications in the balance piston-bearing coolant flow path since the last test series, and these will be discussed below.

During the 1977 test series, a direct measurement of pressure drop across the bearing was not available. This pressure drop was calculated from the balance piston pump pressure measurement and the balance piston return cavity pressure measurement, assuming a slinger pumping coefficient of 0.1. Following this approach, the data indicated that the pressure drop across the bearing at high speeds, approximately 7330 rad/s (70,000 rpm), and nominal balance piston flow was approximately 258 N/cm² (375 psi). Bearing life under the loads due to this pressure drop would be shortened considerably, and it was decided to lower the pressure drop across the bearing. This was accomplished by drilling eight bypass holes of 2.18 mm (0.086 inch) diameter through the bearing cartridge. The duplex bearing with and without bypass was tested in water at zero speed to ensure the proper resistance of the bypass holes. The bearing pressure drop (no bypass) measured at zero speed during the water flow tests approximated that expected based on SSME bearing test experience. Using a nominal SSME bearing dynamic coefficient, $K = 0.65$, provides prediction of pressure drop across the bearing as a function of flow through the bearing at a given speed. The 1977 MK 48 data imply a value of $K = 1.86$, and this was used as a maximum bearing resistance. Expected bearing and bypass pressure drop as a function of flow is shown in Fig. 57 and 58 for pump speeds of 3142 and 7330 rad/s (30,000 and 70,000 rpm), respectively.

Another major area of concern was the capability of the balance piston to provide axial thrust balance. On the previous test series, analysis indicated small margin on the low end of the axial thrust range of the balance piston. Improvement of this margin would be accomplished by lowering the resistance of the flow path downstream of the balance piston. However, the increased balance piston flow due to the lower downstream resistance causes a higher bearing pressure drop as well as lower pump efficiency due to the higher leakage.

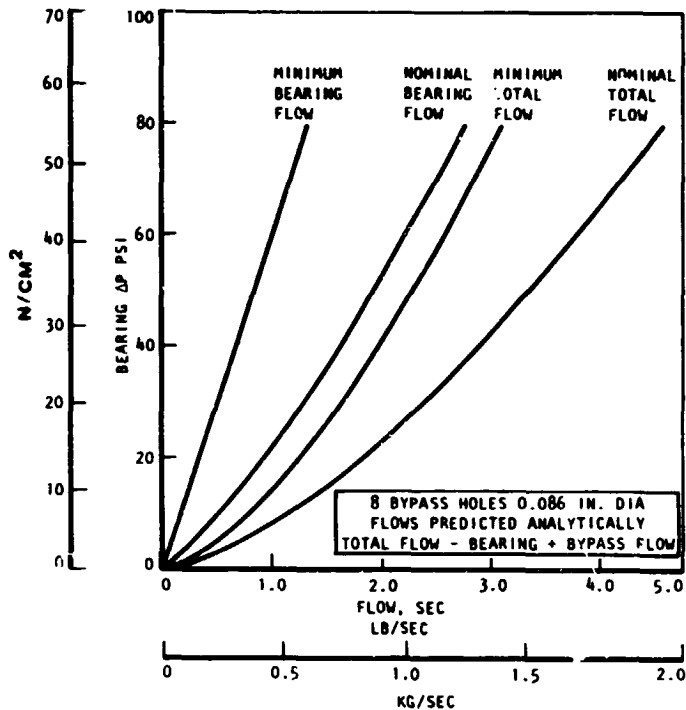


Figure 57. MK 48 LOX Pump (Pump End Bearing Characteristic at 3142 rad/s (30,000 rpm))

A study was conducted to determine a downstream resistance which would keep the balance piston flow as low as possible while still providing adequate capability to balance the axial thrust expected. This condition was determined to be a balance piston flow of approximately 1.2 kg/sec (2.6 lb/sec) at the design point operation. Figure 58 shows that at this flowrate the maximum pressure drop across the bearing would be 62 N/cm² (90 psi), and this provided an acceptable bearing life.

PUMP HYDRODYNAMIC PERFORMANCE

Pump Head Rise

The pump head rise is determined by the relationship:

$$\Delta H = \frac{144}{\rho_{\text{avg}}} \left(P_d - P_i \right) + \frac{V_d^2 - V_i^2}{2g}$$

where P_d and P_i are measured static pressures at the discharge and inlet of the pump, respectively, and V_d and V_i are the average velocities at the discharge and inlet, respectively. These velocities are not measured but are a function only of the measured flowrates and the diameters of the discharge and inlet ducts.

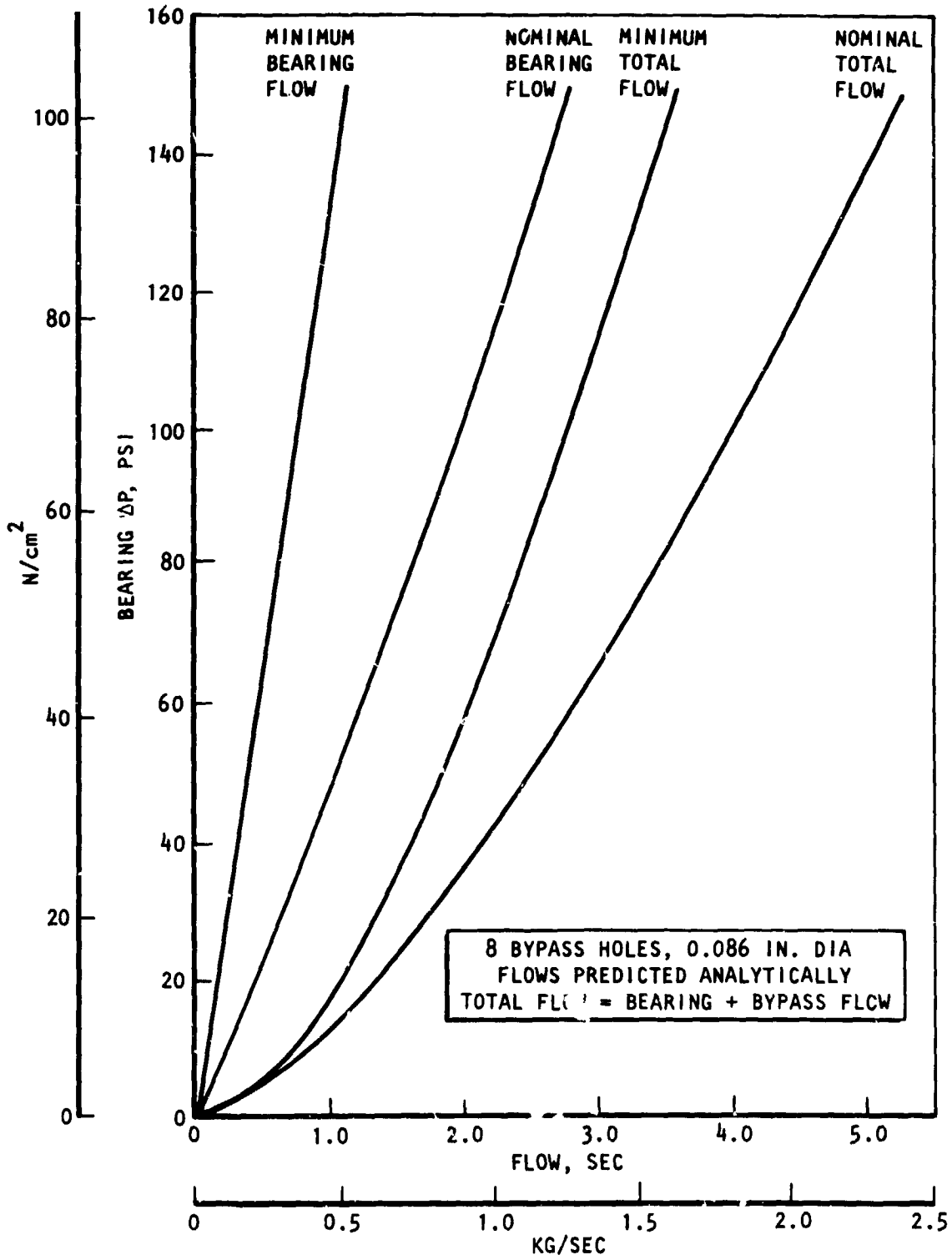


Figure 58. Mark 48 LOX Pump, Pump End Bearing Characteristics at 7330 rad/s (70,000 rpm)

Figure 59 shows the pump overall head rise as a function of flow where both the data and the predicted head are scaled to a speed of 7330 rad/s (70,000 rpm). The scaling was accomplished using the affinity laws which have been thoroughly substantiated as applicable for LOX. The data consist of 36 data points from four tests with test speeds varying from 2995 to 6932 rad/s (28,600 to 66,200 rpm). The symbols used for the data points distinguish the different operating speed ranges tested. The test data cover a range of Q/N from 71% of design Q/N to 109% of design Q/N.

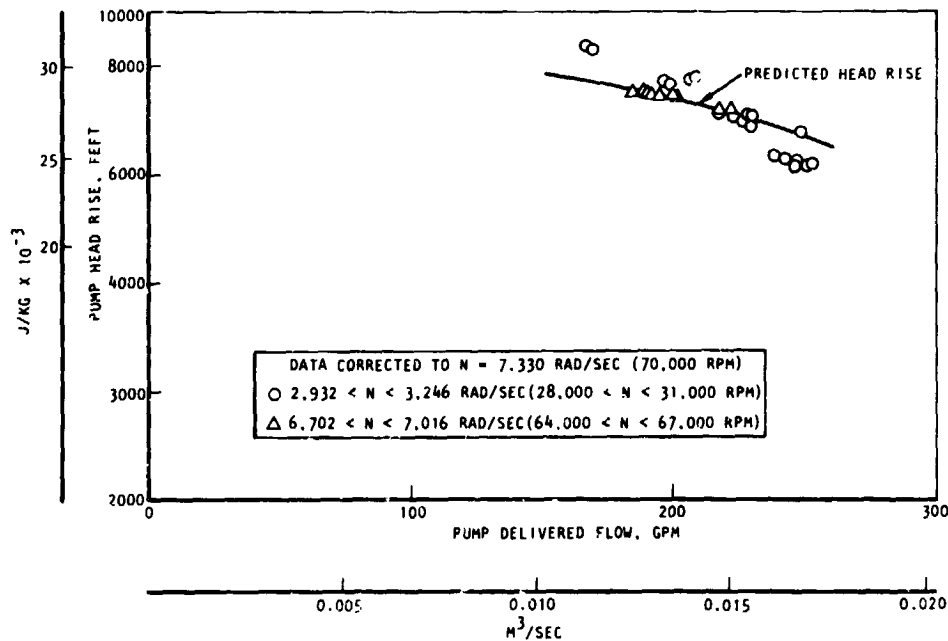


Figure 59. Mark 48 Oxidizer Pump Performance May 1978 Test Series, Pump Head Rise as a Function of Delivered Flow

From Fig. 59, it can be seen that there is agreement between the measured head rise of the tests and the predicted head.

Pump Efficiency

A plot of pump isentropic efficiency as a function of pump delivered flow appears in Fig. 60. The isentropic efficiency is calculated as the fluid horsepower divided by the horsepower achievable, if the fluid were raised from the inlet enthalpy to the exit enthalpy isentropically. Fluid horsepower is based on the inlet and exit pressure of the pump, the average pump density (which does not change significantly for LOX), and the pump-delivered flow. Isentropic horsepower is calculated from the pressures and temperatures at inlet and exit of the pump and the total flow through the pump (delivered flow plus balance piston flow).

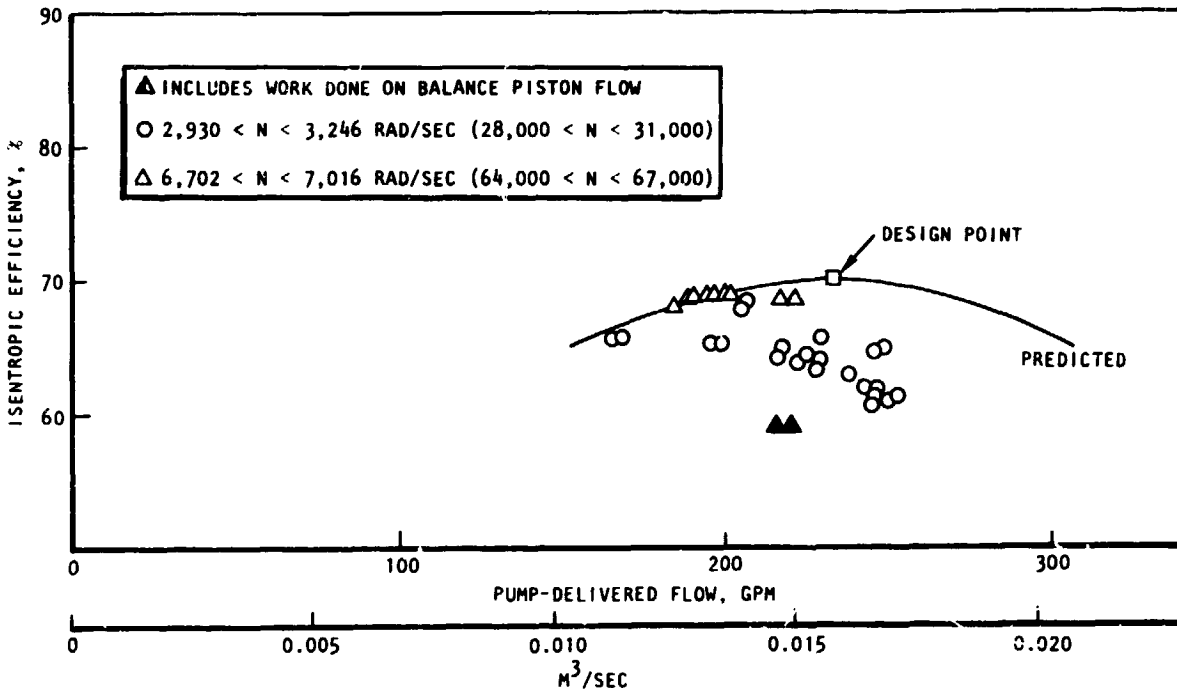


Figure 60. Mark 48 Oxidizer Pump Performance May 1978 Test Series, Isentropic Efficiency as a Function of Delivered Flow Balance Piston Flow Bled Overboard

Figure 60 presents the isentropic efficiency for all data slices of all the tests conducted. At low speeds, the temperature rise from the pump inlet to the pump exit is very small. Any instrumentation errors in the temperature measurements will cause a large error in the isentropic horsepower calculation at low speed. Any instrumentation errors at high speed will cause less error in the isentropic horsepower calculation because of the higher temperature rise at these speeds. Therefore, more confidence is placed upon the data at speeds near 7330 rad/s (70,000 rpm). The isentropic efficiency calculated from the measured parameters at the high speeds appear very near to the predicted efficiency at the design point.

It is observed that the temperature rise of the balance piston flow is considerably higher than that of the pump delivered flow. This indicates additional work being done on that portion of the total flow. To account for this additional work, the isentropic horsepower is calculated as the sum of the pump-delivered flow raised to the energy level based on the pump discharge temperature, and the balance piston flow raised to an energy level based on the balance piston return cavity temperature. The isentropic efficiency calculated in this manner is shown as the closed symbols in Fig. 60. It should be noted that any heat leakage into the balance piston cavity, from outside the pump or from the turbine end causes this calculation of efficiency to be incorrectly low. The values presented in Fig. 60 are thought to be conservative; the true efficiency of the pump being somewhere between the open and closed symbols.

Inducer Static Pressure Rise

The inducer pressure rise is calculated as the difference between the inducer discharge and the pump static inlet pressures. The inducer discharge pressure is measured by a static pressure tap located in the inducer housing (at the inducer tip diameter) just downstream of the inducer. A plot of the inducer pressure rise of the data at approximately 3142 rad/s (30,000 rpm), corrected to 3142 rad/s (30,000 rpm), is shown as a function of pump-delivered flow in Fig. 61. This plot is presented to define the pressure to which future balance piston recirculation flow will exit. The inducer static pressure rise corrected to 7330 rad/s (70,000 rpm) is presented as a function of pump-delivered flow in Fig. 62. All reduced data slices are presented in this plot.

Figure 62 also shows the predicted static pressure rise across the inducer. A comparison between the measured and the predicted pressure rise indicates the slopes to be similar, with the measured data being considerably lower than the pressure rise predicted. It is possible that the static pressure tap located at the tip radius does not correctly measure the average pressure rise for the inducer due to wear ring backflow from the impeller, or for some other reason. However, the measured data indicate a condition which could seriously affect the suction performance of the impeller. This is especially true at the higher flowrates, such as 130% of design flow where the measured data, if extrapolated, would indicate zero static pressure rise across the inducer. Considering this potential problem of the inducer, suction performance tests should be monitored very carefully during the next test series for this pump.

Balance Piston Performance

Balance piston performance was calculated using the Rocketdyne steady-state balance piston performance computer program. The resistance downstream of the balance piston was modeled to account for the bearing, slinger, and overboard bleed line resistance. The impeller discharge pressure measurement was determined to be incorrect. An impeller static pressure rise map generated from the 1977 data was used to predict the impeller pressure at each data slice. The ratio of fluid-to-wheel tangential velocity, designated as K , was assumed to have a value of 0.5. The balance piston program provides performance through a range of axial positions, from the high-pressure orifice full-closed to the low-pressure orifice full-closed. The total balance piston travel, δ , is 0.25 mm (0.01 inch), and eleven equally spaced positions are solved for, $X/\delta = 0.0$ to $X/\delta = 1.0$, where X is the axial opening of the high-pressure orifice. For each position of the balance piston, there corresponds a unique predicted balance piston cavity pressure. The axial position at which the predicted and measured balance piston cavity pressures are equal is considered the match point, or the actual operating position for the particular data slice.

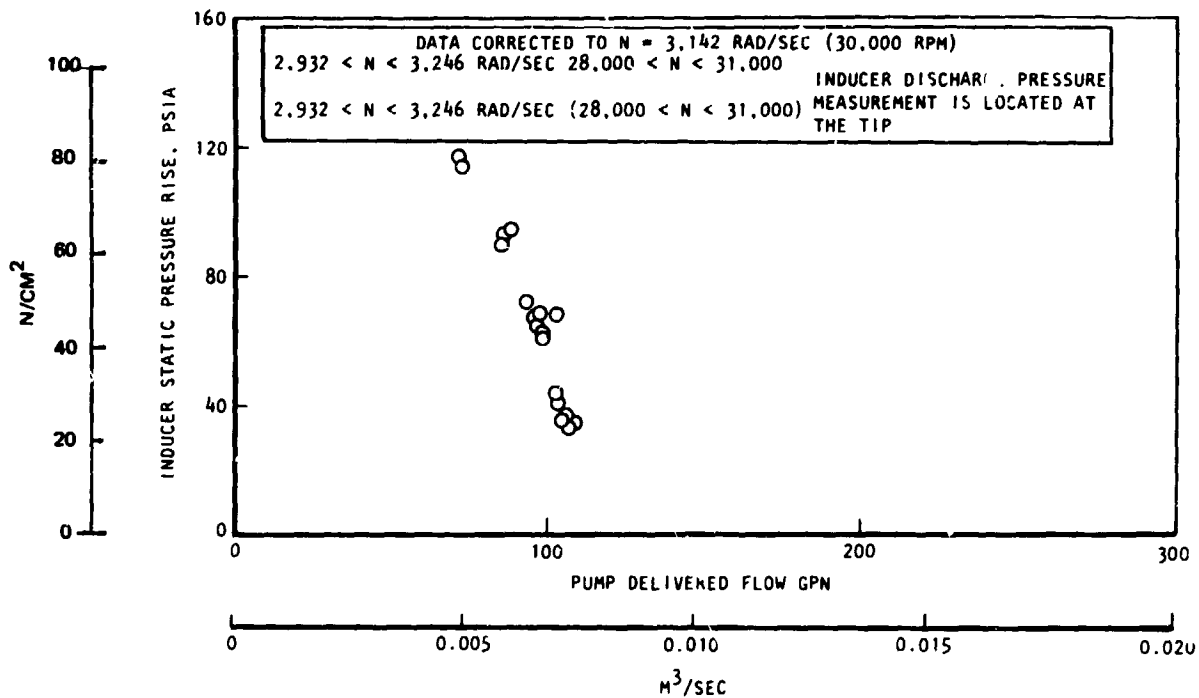


Figure 61. Mark 48 Oxidizer Pump Performance May 1978 Test Series, Inducer Static Pressure Rise as a Function of Delivered Flow

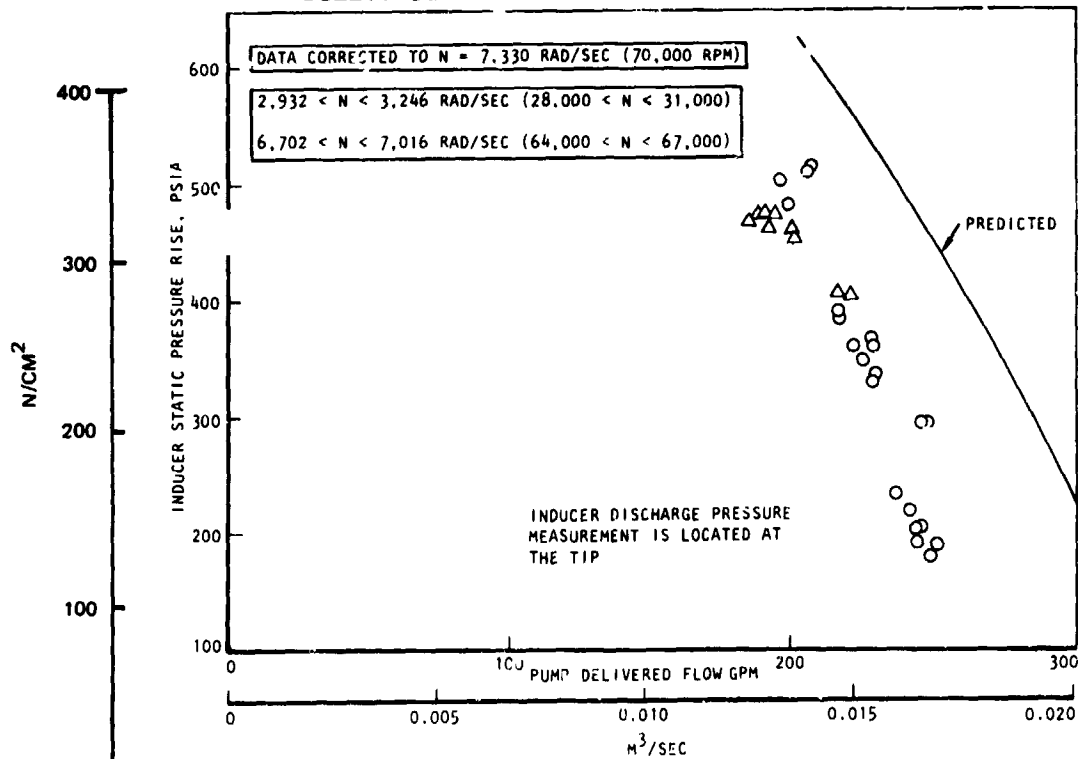


Figure 62. Mark 48 Oxidizer Pump Performance May 1978 Test Series, Inducer Static Pressure Rise as a Function of Delivered Flow

Balance piston performance for Test No. 006, which contains test speeds near 7330 rad/s (70,000 rpm), is presented in Table 14. Parameters presented in Table 14 are:

Pump speed, N

Pump flow/pump speed ratioed to design pump flow/design pump speed

Ratio of high-pressure orifice axial opening to total balance piston travel, X/δ

Balance piston flowrate at match point (from balance piston program)

Balance piston flowrate measured with an orifice in the overboard bleed line

Balance piston thrust at match point, F_{match}

Thrust achievable with low-pressure orifice closed, F_{max} , minus thrust achievable with high-pressure orifice closed F_{min}

F_{max} minus F_{match}

$F_{\text{match}} - F_{\text{min}}$

Scaled thrust range $K_T = \frac{F_{\text{max}} - F_{\text{min}}}{N^2}$ scaled to 7330 rad/sec (70,000 rpm)

Balance piston sump pressure predicted (from balance piston program)

Measured balance piston sump pressure

The predicted balance piston flow (Table 14) agrees very well with the overboard bleed measured flow. The largest difference between the two flows is 3.3% at low speed and 2.5% at high speed. Another indication of the validity of the model is the agreement between the predicted and measured balance piston sump pressure. These are presented in the last two columns of Table 14; the maximum difference between them is 5%.

The thrust range of the balance piston is adequate. That range is the difference between the maximum balance piston thrust (high-pressure orifice full-open) and the minimum thrust (high-pressure orifice full-closed), and is presented as $F_{\text{max}} - F_{\text{min}}$ in Table 14. The thrust range coefficient labeled K_T presents the thrust range scaled to 7330 rad/s (70,000 rpm), assuming the thrust range can be scaled by the speed squared.

Ideally, the preferred operating point of the balance piston would be directly in the middle of the available thrust range. The thrust at the match point (Table 14) is closer to the maximum thrust than the minimum thrust. At low speed the margin available on the maximum balance piston thrust is approximately 16% of the thrust range. High-speed data show the margin to be 26 to 32% of the thrust range.

TABLE 14. MK 48 OXIDIZER BALANCE PISTON PERFORMANCE TEST 006, 5/31/78

SLICE	N (RPM)	C/N (Q/N) DESIGN	X/6	\dot{W} PREDICTED (LB/SEC)	\dot{W} MEASURED WITH ORIFICE (LB/SEC)	\dot{F} MATCH (LB)	$(F_{MAX}-F_{MIN})$ (LB)	$(F_{MAX}-F_{MATCH})$ (LB)	$(F_{MATCH}-F_{MIN})$ (LB)	K_T (LB)	P_{SUMP} PREDICTED (PSIA)	P_{SUMP} MEASURED (PSIA)
1	29909	0.988	0.466	1.674	1.712	2870	2198	351	1847	12040	387.0	368.3
2	29568	1.064	0.470	1.635	1.690	2758	2100	331	1769	11770	370.1	362.2
3	29689	1.069	0.468	1.639	1.692	2764	2108	337	1771	11719	372.3	363.3
4	22748	1.069	0.460	1.643	1.691	2758	2110	350	1760	11683	376.3	366.5
5	64519	0.791	0.330	2.705	2.663	8117	7162	1855	5307	8431	1266.2	1206.4
6	65035	0.802	0.322	2.728	2.683	8149	7286	1956	5330	8441	1268.0	1220.6
7	65176	0.824	0.323	2.727	2.688	8158	7310	1981	5329	8432	1271.1	1222.7
8	65141	0.811	0.313	2.715	2.666	8140	7316	2025	5291	8448	1275.0	1232.6
9	65504	0.814	0.312	2.733	2.666	8203	7413	2065	5348	P166	1283.6	1252.4
10	65657	0.837	0.304	2.725	2.668	8155	7417	2128	5289	8431	1273.5	1251.7
11	65833	0.861	0.304	2.710	2.655	8188	7401	2100	5301	8368	1284.4	1263.6
12	66032	0.863	0.300	2.709	2.657	8173	7427	2147	5280	8346	1280.3	1268.6
13	66016	0.867	0.301	2.705	2.648	8193	7421	2128	5293	8321	1285.8	1275.2
14	66032	0.933	0.285	2.629	2.608	7799	7733	2302	4931	8128	1243.6	1260.0
15	66163	0.951	0.280	2.613	2.606	7758	7234	2354	4880	8097	1244.7	1265.9

Pump End Bearing

Direct measurements of the pressure drop across the pump end bearing were available for this test series. During Tests 003 and 004, individual pressure transducers were used to measure the pressure upstream and downstream of the bearing. During Tests 005 and 006, an additional differential pressure transducer was installed between the bearing upstream and downstream instrumentation lines. The pressure drop across the pump end bearing is presented as a function of speed (Fig. 63). There is a large data scatter (Fig. 63) even when a differential pressure transducer is used. The pressure drop across the bearing is proportional to the flow squared, and if flow were varying significantly, a large variation in pressure drop could be expected. However, measured balance piston flow as a function of speed (Fig. 64) is seen to be consistent and, therefore, should not be the cause of the bearing pressure drop data scatter.

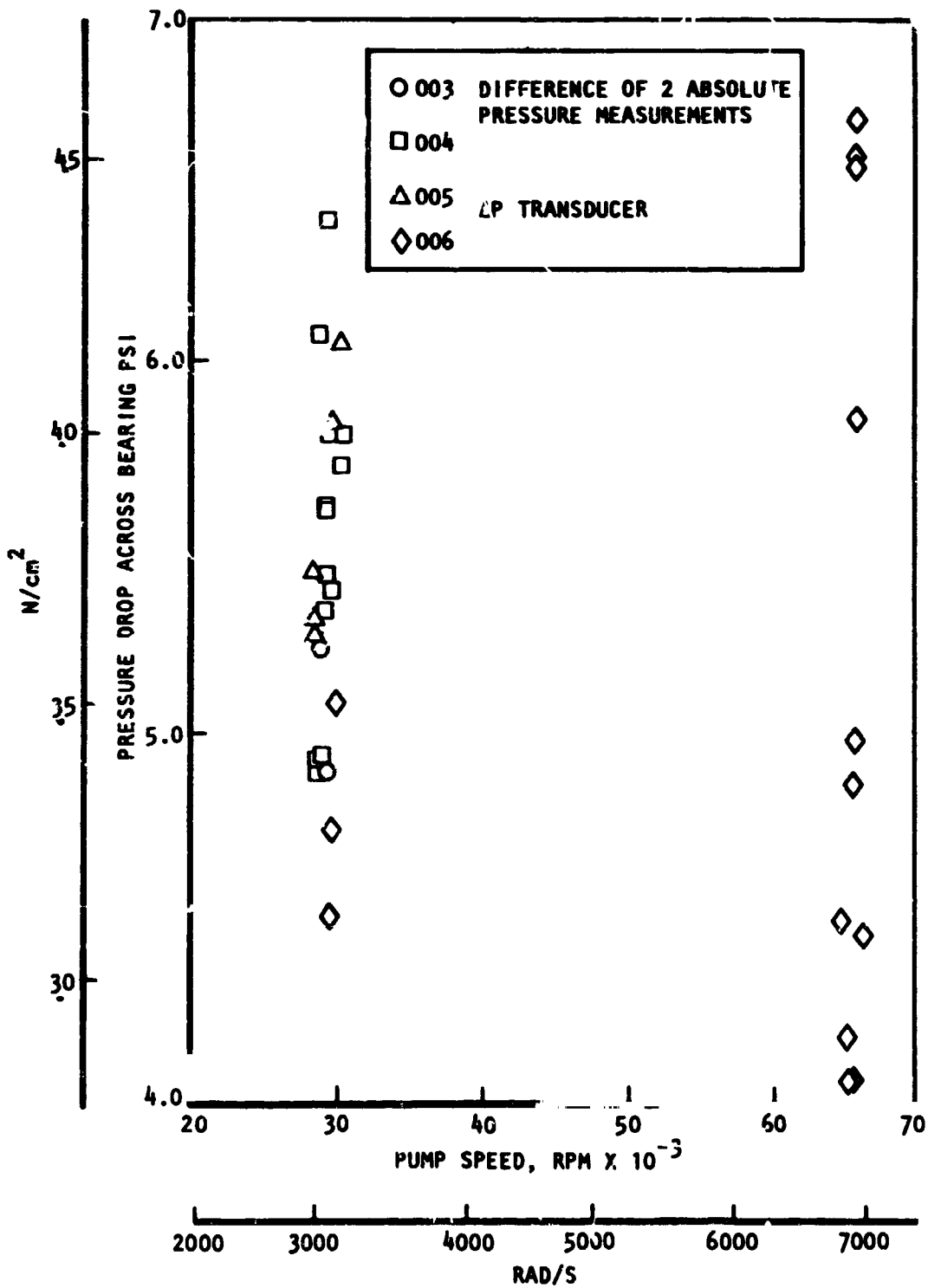
Another problem presented by the data is that of determining the true bearing resistance (with bypass). Assuming the value for C_0 of 0.578 as obtained during the water-flow calibration of the bearing to be correct, as well as the bypass hole resistance, a K value to be used in the bearing pressure drop equation can be calculated from the data. The low-speed data indicate K values ranging from 5.14 to 39.79, with a mean value of $K = 18.07$. The range of the K value for the high-speed data is 0.41 to 1.03, the average being 0.66. An error in either the bearing upstream or downstream pressure measurements may be postulated to explain the data, but as of this writing there is no basis for this. However, the value that matches the high-speed data does show good agreement with values obtained for SSME bearings. The values derived from the low-speed data not only disagree with the SSME bearing data, but also disagree with theoretical considerations which led to the bearing resistance equation used.

Balance Piston Flow Temperature Rise

It was found at high speed operation that the temperature of the fluid in the balance piston return cavity was higher than anticipated. This is of concern since the balance piston flow is being heated by, among other components, the pump end bearing. High-measured temperature of this flow could be the result of a bearing failure or bearing distress. (It should be noted that the bearings were in good condition after pump disassembly.) However, operation at high speeds was limited to approximately 8 seconds. The temperature rise from the pump discharge to the balance piston return cavity is presented as a function of pump speed in Fig. 65. What appears to be scatter in the data at the high speed is actually due to the response characteristic of the temperature bulb instrumentation as the speed ramps. The highest value of 79 degrees temperature rise is at a point in which the temperature has become steady.

The heat input to the balance piston flow originates from the following sources. There is some heat conducted from any areas of the pump structure with higher temperature than that of the fluid, as well as heat conducted through the pump insulation. This heat input should be minimal and is not a

C-2



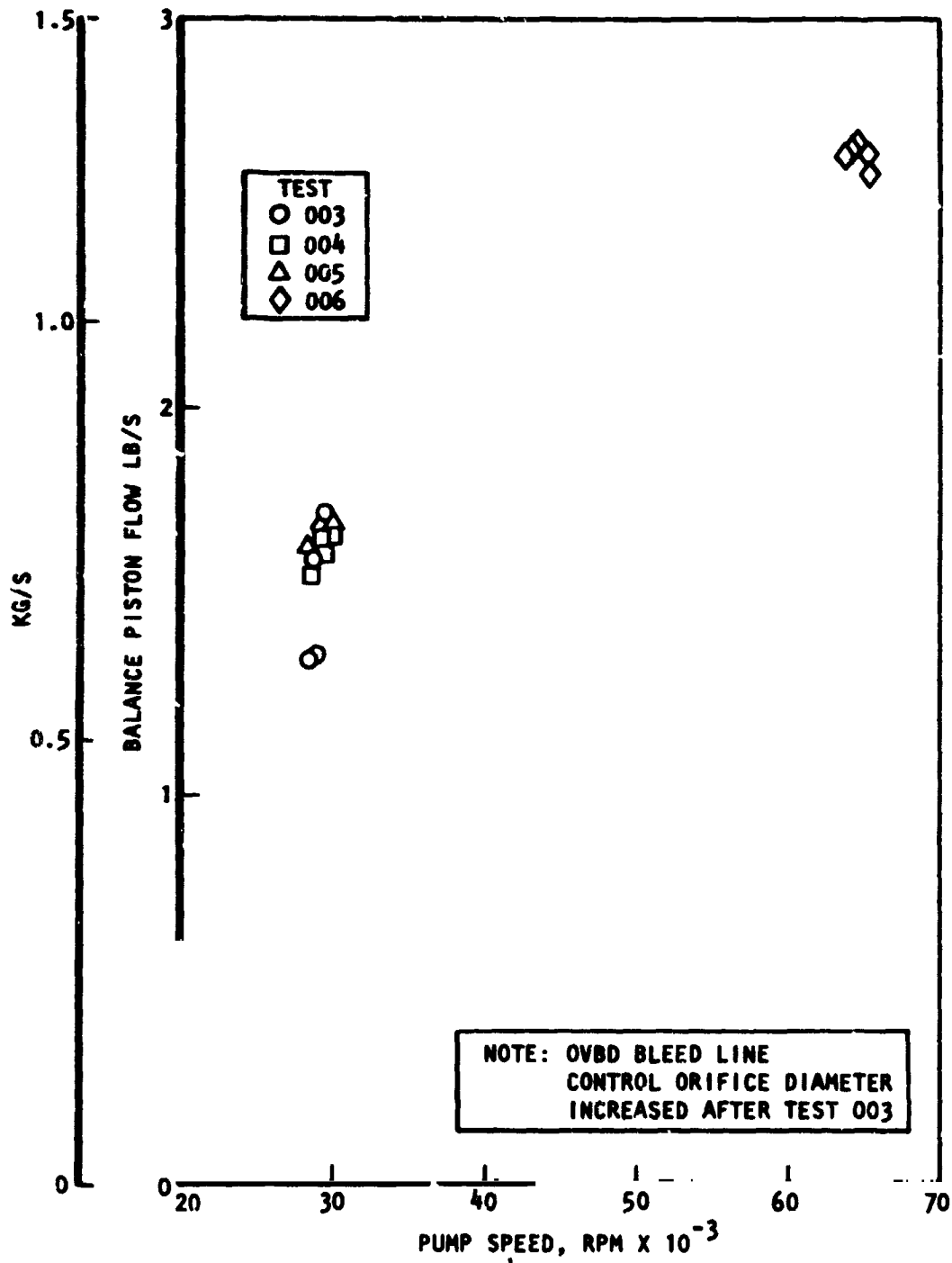


Figure 64. Mark 48 Oxidizer May 1978 Test Series, Balance Piston Measured Flow as a Function of Pump Speed

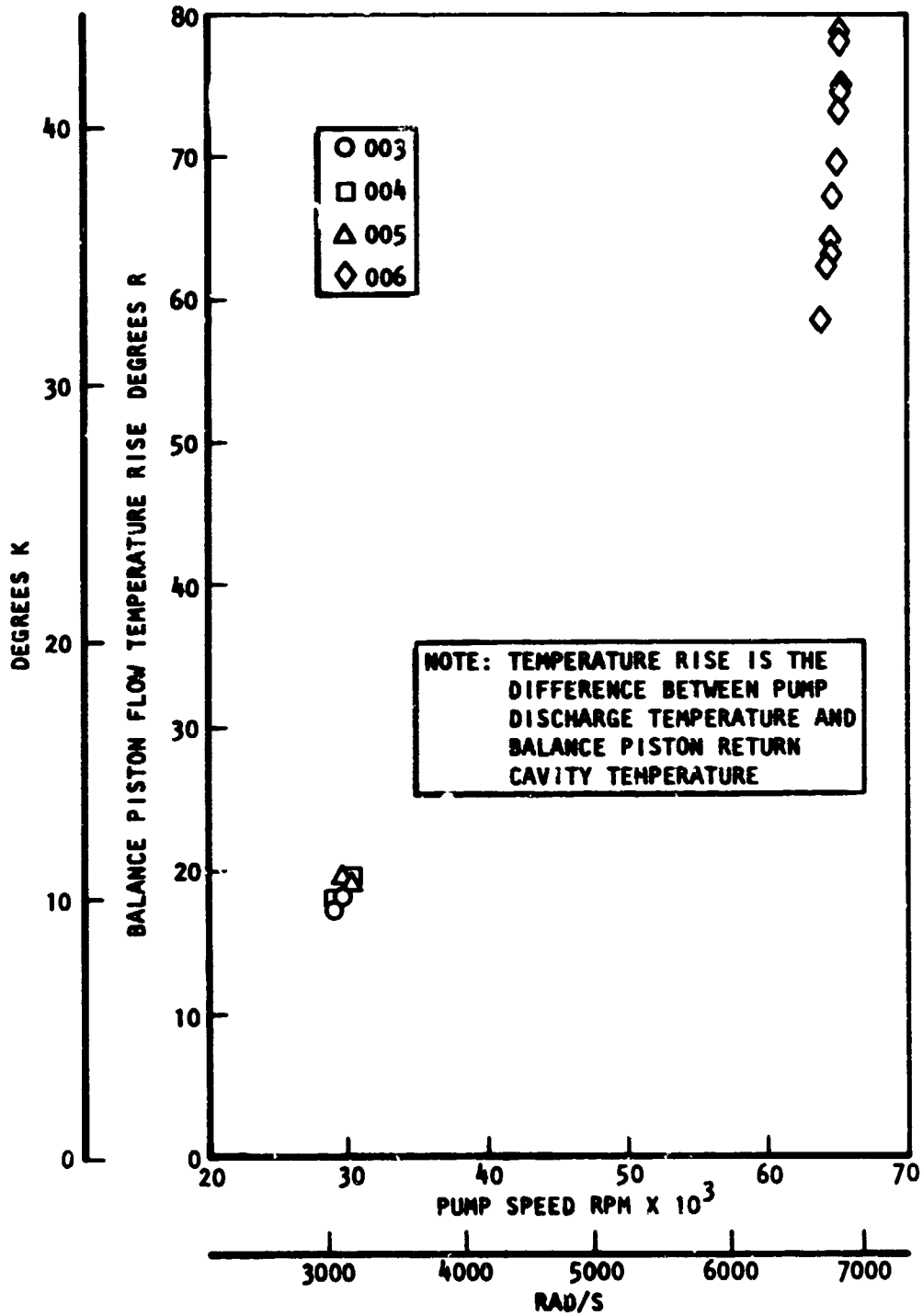


Figure 65. Mark 48 Oxidizer May 1978 Test Series, Balance Piston Flow Temperature Rise as a Function of Pump Speed

strong function of pump speed so was not considered in the analysis to be presented. The balance piston as well as the front face of the slinger generate heat which is absorbed by the fluid. The slinger back surface contains ribs for the purpose of lowering the pressure of the seal leakage flow. This ribbed face adds considerable heat to the fluid. Some of this heat is carried away in the seal leakage flow, but most of it is carried back into the cavity area because the recirculation flow is expected to be much larger than the leakage flow. Lastly, the bearing inputs heat to the balance piston flow.

A study was performed to predict the balance piston flow temperature rise measured at the high speed, specifically 93.9 K (79 R) at 6929 rad/s (66,163 rpm). Nominal empirical heating coefficients for each of the heat-generating components were increased individually to a value which predicted the measured temperature rise. The nominal values as well as the values of each coefficient which resulted in matching the measured temperature rise are presented in Table 15. For the bearing to be producing the heat required to match the measured temperature rise (see Table 15), it would require approximately 56 times the nominally predicted heat input for such a bearing. Considering that the bearing was in good condition posttest, this is very unreasonable.

TABLE 15. MK 48 OXIDIZER HEATING COEFFICIENTS
AFFECTING BALANCE PISTON FLOW

	Nominal	Required to Match Data	Increase
Bearing Heat Input	0.8 Btu/sec	44.6	55.8
Balance Piston Torque Coefficient	$0.04/R_e^{1/5}$	$0.259/R_e^{1/5}$	6.5
Slinger Front Face Torque Coefficient	$0.04/R_e^{1/5}$	$0.612/R_e^{1/5}$	15.3
Slinger Ribbed Back Face Torque Coefficient	0.01	0.0376	3.8

NOTE: $R_e = \frac{U_{TIP} R_{TIP}}{\nu}$

The heating coefficient requiring the least change to predict the measured temperature rise is that of the slinger-ribbed back face. Experimental values reported in Ref. 2 show this torque coefficient's maximum value to be approximately 0.02.

It is desirable to reduce the temperature of this return flow because eventually this fluid will be returned to the impeller eye, and a lower temperature will improve suction performance. Based on the above analysis, the best way to lower the temperature is to reduce the effect of the heating by the slinger which can be accomplished by trimming the slinger outer diameter. Testing with a trimmed diameter should also verify that this is indeed the major source of the high temperature. A study was performed to evaluate the effect of the trim on those parameters dependent upon it.

It was assumed that the bearing, balance piston, and slinger front face contribute nominal heat inputs, while the ribbed face of the slinger is actually contributing with a torque coefficient of 0.0376. Although it is probable that the other coefficients are somewhat higher than nominal, it can be seen that the back of the slinger is the major component of the overall heat input.

With the heating coefficients set in this manner, an analysis was made of the effect of slinger height on the balance piston flow temperature rise, axial thrust, and sealing performance.

The balance piston flow temperature rise as a function of slinger height is shown in Fig. 66. The decreasing slope of the temperature rise as the radius is increased is due to changes in fluid properties with temperature change. The effect of slinger height on the net slinger axial thrust is shown in Fig. 67. Figures 68 and 69 show the effects of slinger height on vaporization of the fluid and, therefore, sealing performance of the slinger. Figure 68 shows the radius at which the vapor pressure is reached as a function of slinger tip radius. It can be seen that for a slinger tip radius of approximately 24.8 mm (0.975 inches), vaporization occurs just at the seal radius. Slinger height below this radius will result in liquid at the seal with a potential increase in seal leakage. Figure 69 shows the pressure expected at the seal as a function of the slinger height. The discontinuity in the curve is at the slinger height at which the predicted vapor pressure is not reached at the seal radius.

It is recommended that the slinger tip radius not be reduced beyond 25.4 mm (1.0 inch), for the purpose of maintaining vapor at the seal radius. The K value assumed for the ribbed surface of the slinger is 0.7. It is felt that this is conservative and, therefore, ensures vapor at the seal radius.

Slinger Front-Face Pumping Effectiveness

In addition to lowering the pressure and vaporizing the fluid entering the seal, the slinger was designed to decrease the pressure downstream of the balance piston low-pressure orifice. This is accomplished by the pumping action on the front face of the slinger which occurs by adding energy to the flow with some percentage of the slinger tangential speed imparted to the fluid. This average tangential velocity was assumed initially to be one-half the wheel tangential velocity which corresponds to a value of $K = 0.5$. However, as the axial gap between the slinger front face and the stationary face increases, it is expected that the pumping effectiveness should decrease.

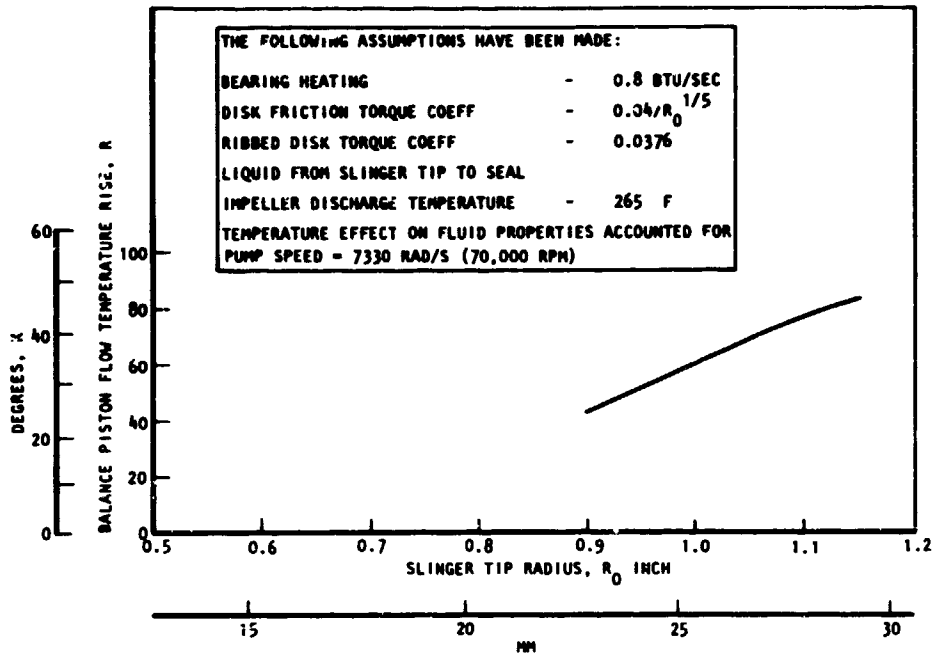


Figure 66. Mark 43 Oxidizer Expected Balance Piston Temperature Rise as a Function of Slinger Height

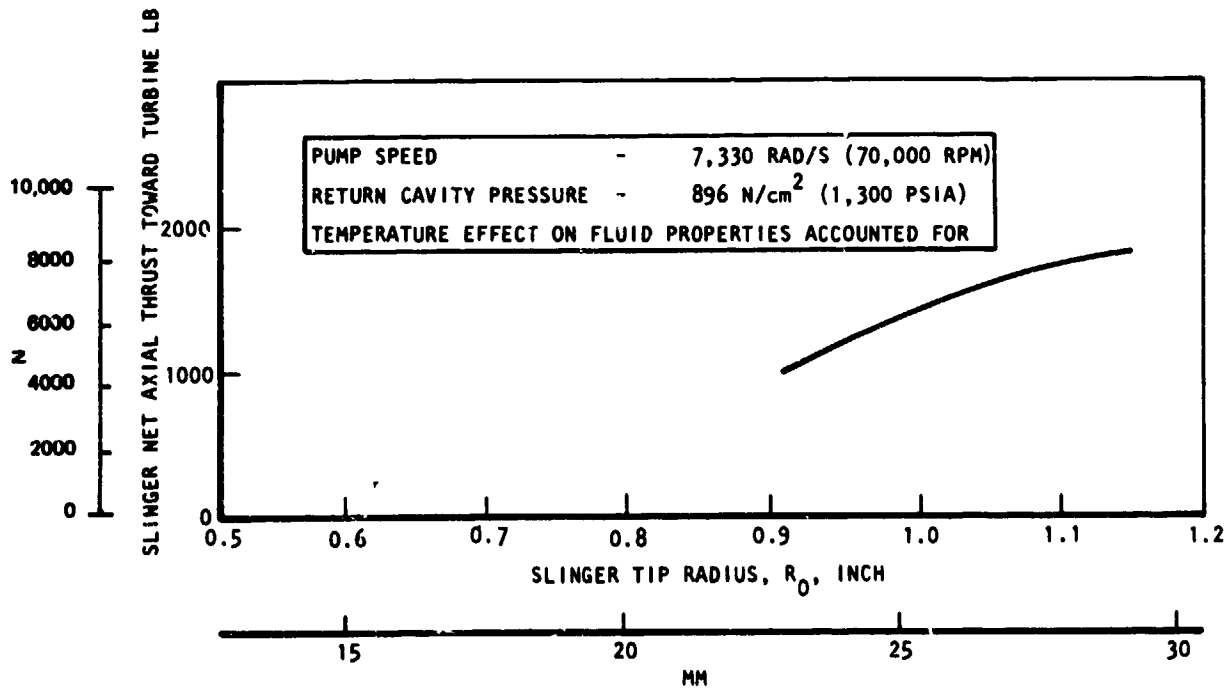


Figure 67. Mark 48 Oxidizer Expected Net Slinger Thrust as a Function of Slinger Height

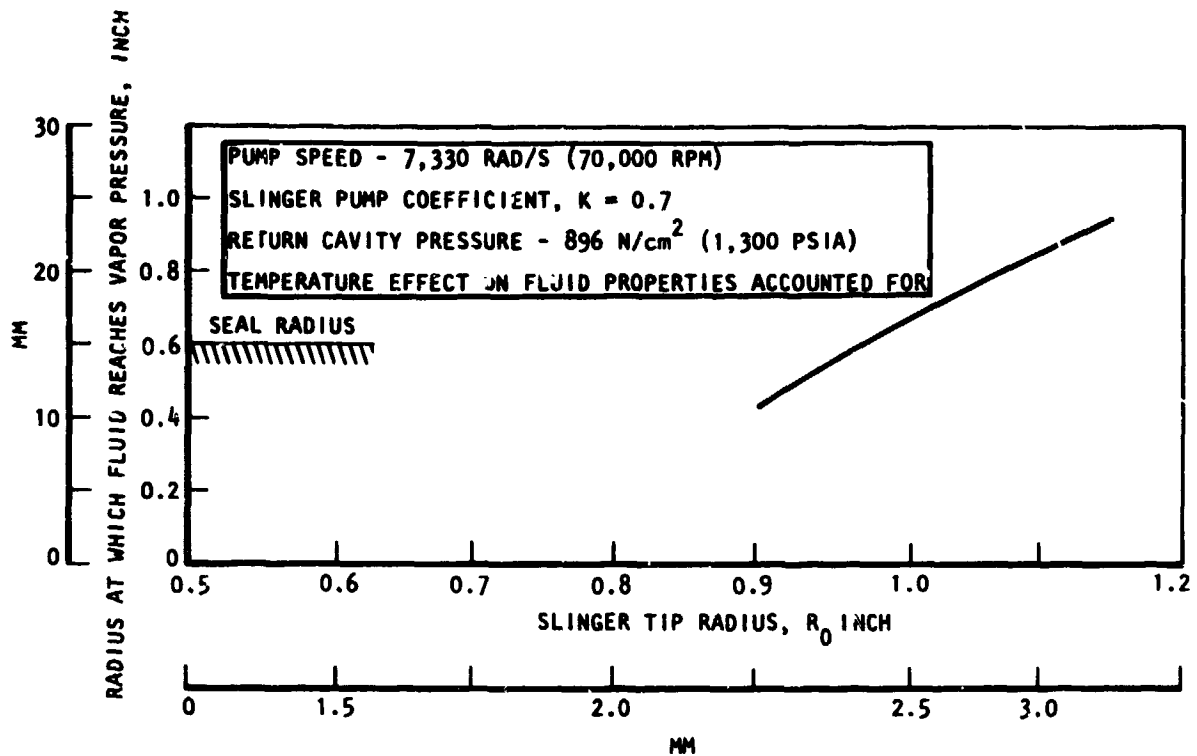


Figure 68. Mark 48 Oxidizer Expected Radius of Vapor Interface on Back of Slinger as a Function of Slinger Height

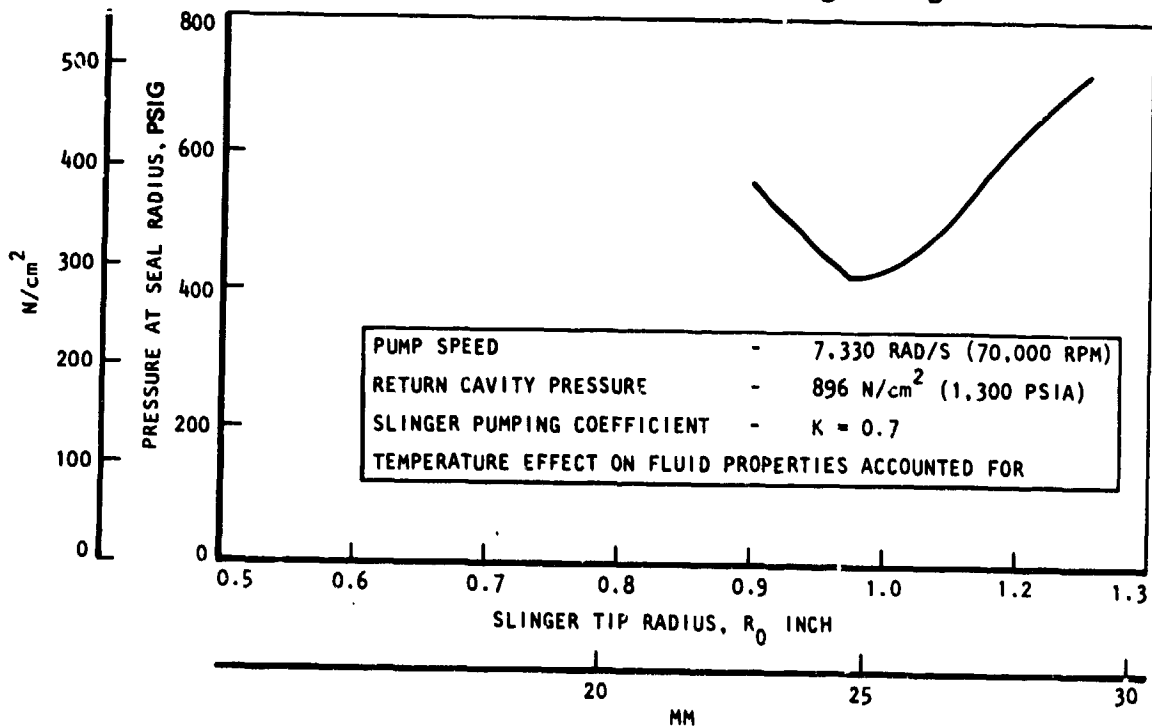


Figure 69. Expected Seal Pressure as a Function of Slinger Height Assuming Vaporization of Pumped Fluid when Pressure Reaches Vapor Pressure

Pressure measurements downstream of the bearing and in the balance piston return cavity provide the upstream and downstream pressure measurements, respectively, across the slinger. Based on these pressure measurements a slinger K value was calculated for both low- and high-speed data, and these values are presented in Table 16. Low speed data (Table 16) indicate a slinger K value of 0.17, whereas the high speed data indicate a lower value of approximately $K = 0.05$ with a considerable scatter in this calculated quantity. High speed data are thought to be more accurate since the pressure levels are nearer the mid-range of the transducers. These data show a very ineffective pumping occurring due to the front face of the slinger.

TABLE 16. MK 48 OXIDIZER SLINGER PUMPING EFFECTIVENESS
TEST 006, 5/31/78

Slice	Pump Speed,		Slinger Pumping Coefficient, K
	rad/s	(rpm)	
1	3132	(29909)	0.19
2	3096	(29568)	0.18
3	3109	(29689)	0.14
4	3115	(29748)	0.16
5	6756	(64519)	0.00
6	6810	(65035)	0.00
7	6825	(65176)	0.05
8	6822	(65141)	0.00
9	6860	(65504)	0.00
10	6980	(66657)	0.07
11	6894	(65833)	0.10
12	6915	(66032)	0.11
13	6913	(66016)	0.10
14	6915	(66032)	0.11
15	6929	(66163)	0.04

MECHANICAL PERFORMANCE

The second test series revealed a mechanical problem with the primary seal retaining nut which had the tendency to back off as a result of the influence of a rotating fluid field around it. This led to blockage of the balance piston drain cavity ports, and eventually to loss of axial thrust control and pump damage. As described in a prior section, the nut and its lock were re-designed, and the new configuration was evaluated in the third test series, in 1978. The test experience with the modified nut indicated that the problem of backing off has been corrected.

One other minor problem of a mechanical nature has been observed regarding the chromium plating on the rotor under the primary hot-gas turbine seal. Chrome plating applied to the rotor flaked off, possibly contributing to the high torque observed in the May 1978 test series. The cause of the flaking was identified as a sharp corner where the plating terminated which resulted in inadequate adherence and eventually led to chipping and flaking. The condition will be corrected on future builds by eliminating the sharp transition in the rotor before plating is applied.

The condition of the turbine end bearings was excellent after each test series, even after having gone through a failure on the pump end. No indication of excessive radial or axial loading, overheating or distress of any kind was observed. The pump end bearings were largely consumed in the fire during the 1977 test series and, therefore, their pre-failure condition is unknown. The pump end bearings were in excellent condition after the third (May 1978) test series. Posttest analysis indicated adequate cooling and low-coolant-pressure differential. The nominal axial load was approximately 445 N (100 lb) on each bearing with a maximum excursion to 1023 N (230 lb). The radial load on the No. 1 bearing (numbered from impeller side) was zero as intended. The radial load on the No. 2 bearing was 445 N (100 lb). Thus, based on the condition of the bearings after this test series, the bearings were functioning properly.

The performance of all four shaft dynamic seals was excellent in all tests. Pressure levels in the drain systems were maintained at sufficiently low levels to preclude intermixing of the pump and turbine propellants. The primary LOX seal in particular has been proven a very reliable, rugged concept. In conjunction with the slinger, its leakage rate at design speed was approximately 0.068 kg/s (0.15 lb/sec).

APPENDIX A
DESIGN GROUND
RULES

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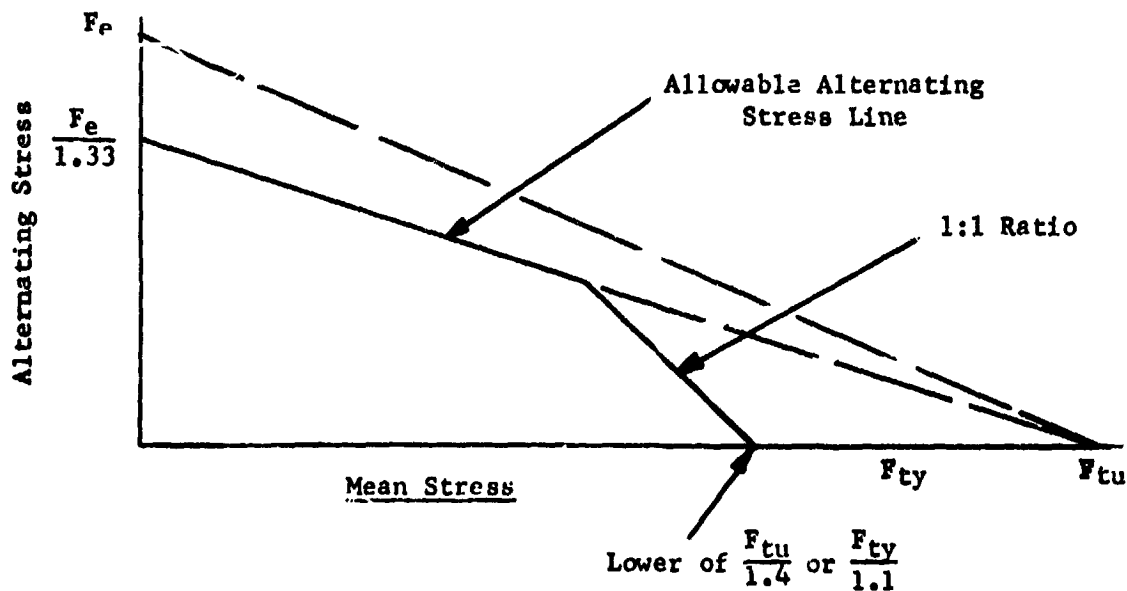
APPENDIX A
DESIGN GROUND RULES

General

Components which are subject to a low cycle fatigue mode of failure shall be designed for a minimum of 300 cycles times a safety factor of 4.

Components which are subject to a fracture mode of failure shall be designed for a minimum of 300 cycles times a safety factor of 4.

Components which are subject to a high cycle fatigue mode of failure shall be designed within the allowable stress range diagram (based on the material endurance limit). If stress range material property data are not available, modified Goodman diagrams constructed as shown below shall be utilized.



F_e = Material Endurance Limit
 F_{ty} = Material Yield Strength (.2% offset)
 F_{tu} = Material Ultimate Strength

APPENDIX A

Effective stress shall be based on the Mises-Hencky constant energy of distortion theory.

Unless otherwise noted under component ground rules specified herein, the following minimum factors of safety shall be utilized:

Factor of Safety (.2% yield) = $1.1 \times \text{Limit Load}$

Factor of Safety (Ultimate) = $1.4 \times \text{Limit Load}$

Limit Load: The maximum predicted load or pressure at the most critical operating condition

Components subject to pressure loading shall be designed to the following minimum proof and burst pressures:

Proof Pressure = $1.2 \times \text{Limit Pressure}$

Burst Pressure = $1.5 \times \text{Limit Pressure}$

Impeller

Inducers and/or impellers utilized in the high pressure pumps shall be designed for operation above incipient cavitation.

Impeller burst speed shall be at least 20% above the maximum operating speed.

Impeller effective stress at 5% above the maximum operating speed shall not exceed the allowable .2% yield stress. (Does not apply to areas in which local yielding is permitted.)

Turbine

Blade root steady-state stress shall not exceed the allowable 1% ten hour creep stress.

Stress state at the blade root as defined by the steady-state stress and an assumed vibratory stress equal to the gas bending stress shall be within the allowable stress range diagram or modified Goodman diagram.

No blade natural frequencies within $\pm 15\%$ of known sources of excitation at steady-state operating speeds.

Disk burst speed shall be at least 20% above the maximum operating speed.

APPENDIX A

Disk maximum effective stress at 5% above the maximum operating speed shall not exceed the allowable .2% yield stress. (Does not apply to areas in which local yielding is permitted).

Bearings

Turbopump designs shall utilize ball bearings.

Maximum DN: 1.5×10^6

B₁₀ life 100 hours

Material:

Rolling Elements	440C
Races	440C

Seals

Turbopump designs shall utilize conventional type seals. However, provision shall be made in the design to permit the incorporation (retrofit) of controlled fluid film (hydrodynamic) face seals. Any rework or modification of the turbopump housing or other component parts in the area of the seals to accommodate the hydrodynamic seals shall be specified. Such modifications should be kept to a minimum.

Face contact seal maximum PV, FV, and P_fV factors:*

	<u>LO₂</u>	<u>H₂+H₂O</u>
PV Factor	25,000	10,000
FV Factor	2,000	800
P _f V Factor	60,000	20,000

*PV = unit load times rubbing velocity (lb/in² x ft/sec)

FV = face load per unit length times rubbing velocity (lb/in x ft/sec)

P_fV = fluid pressure differential times rubbing velocity (psig x ft/sec)

Critical Speed

Rotor bending frequency shall be at least 25% above the rotor maximum operating speed.

A minimum margin of 20% shall be maintained between rotor rigid body critical speeds and rotor steady-state operating speeds at full thrust and the pumped-idle thrust condition. Rigid body critical speeds within the throttled-to-full thrust range shall be permitted only if deemed necessary by both the Contractor Program Manager and the NASA Project Engineer.

APPENDIX B

MARK 48-0 TURBOPUMP ASSEMBLY
DRAWING RS009820E

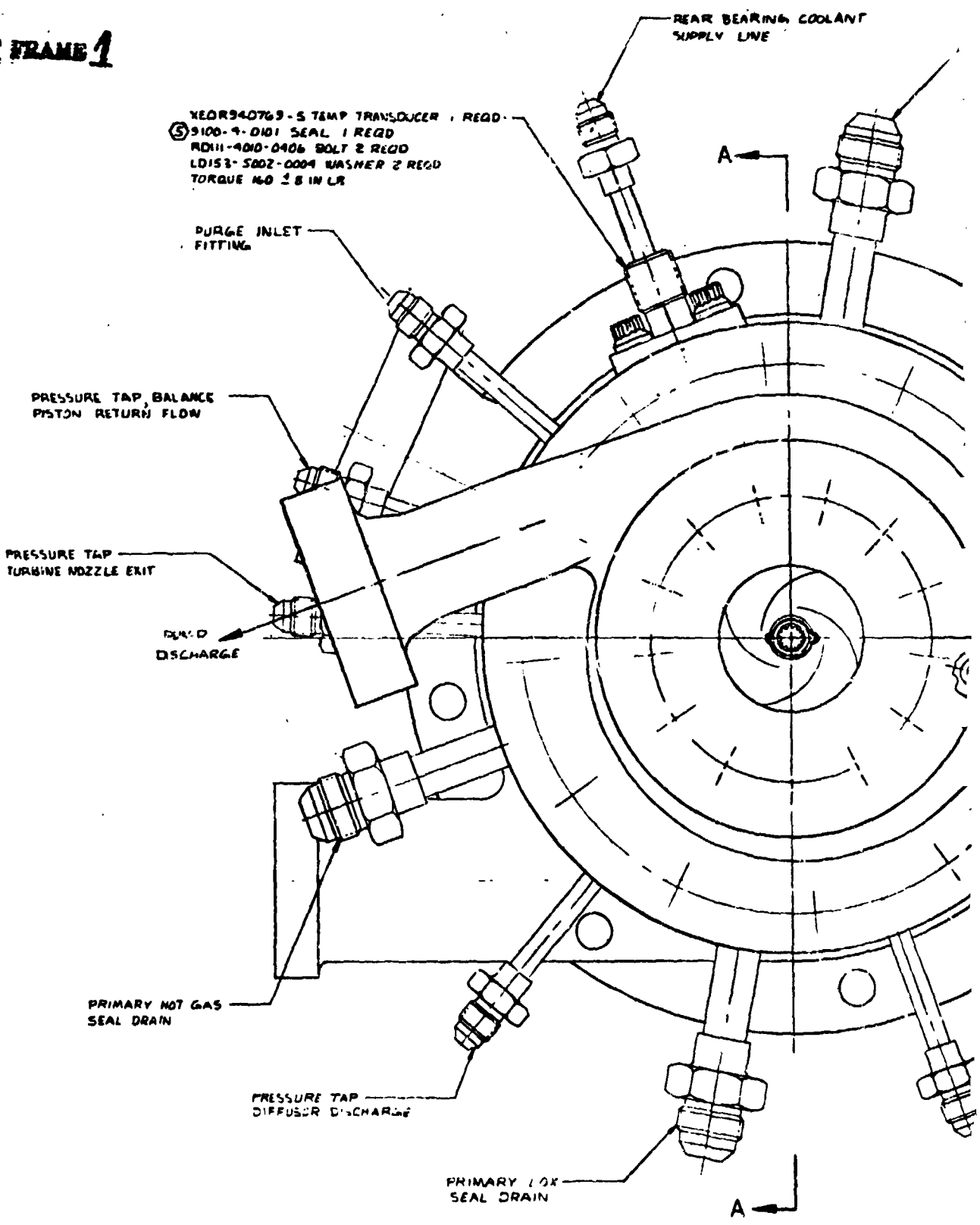
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REPRODUCTION

H
G
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E
D
C
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A

FOLDOUT FRAME 1

- YEDR940769 - 5 TEMP TRANSDUCER 1 REQD.
- ⑤ 9100-9-0101 SEAL 1 REQD
- RD11-4010-0406 BOLT 2 REQD
- LD153-5002-0004 WASHER 2 REQD
- TORQUE NO 28 IN LB



24 23 22 21 20

FOLDOUT FRAME *a*

PRIMARY LOX SEAL
DRAIN, 2 PLACES, 180° PART

a

SECONDARY HOT GAS
SEAL DRAIN

PRIMARY HOT GAS
SEAL DRAIN

PRESSURE TAP
BALANCE PISTON
SUMP

⑤ 9100-4-D101 SEAL 4 REQD

PRESSURE TAP
BALANCE PISTON
CAVITY

PRESSURE TAP
IMPELLER DISCHARGE

R5009612E SPRING 1 REQD

R5009612E SPRING 1 REQD

R5009616E SPACER 2 REQD

R5009644E-003 IMPELLER 1 REQD

SEE VIEW B

⑩ 99R5009647-003 NUT 1 REQD
99R5009603-003 LOCK 1 REQD
TORQUE 650 ± 25 IN LB

⑪ R5009650E-003 INDUCER 1 REQD

⑬ FD111-4010-5422 BOLT 13 REQD
RD133-5002-0004 WASHER 13 REQD
TORQUE 70 ± 3 IN LB

⑭ 99R50098C1-003 BOLT 1 REQD
99R5009202-003 LOCK 1 REQD
TORQUE 75 ± 5 IN LB

R5009643E-003 DIFF

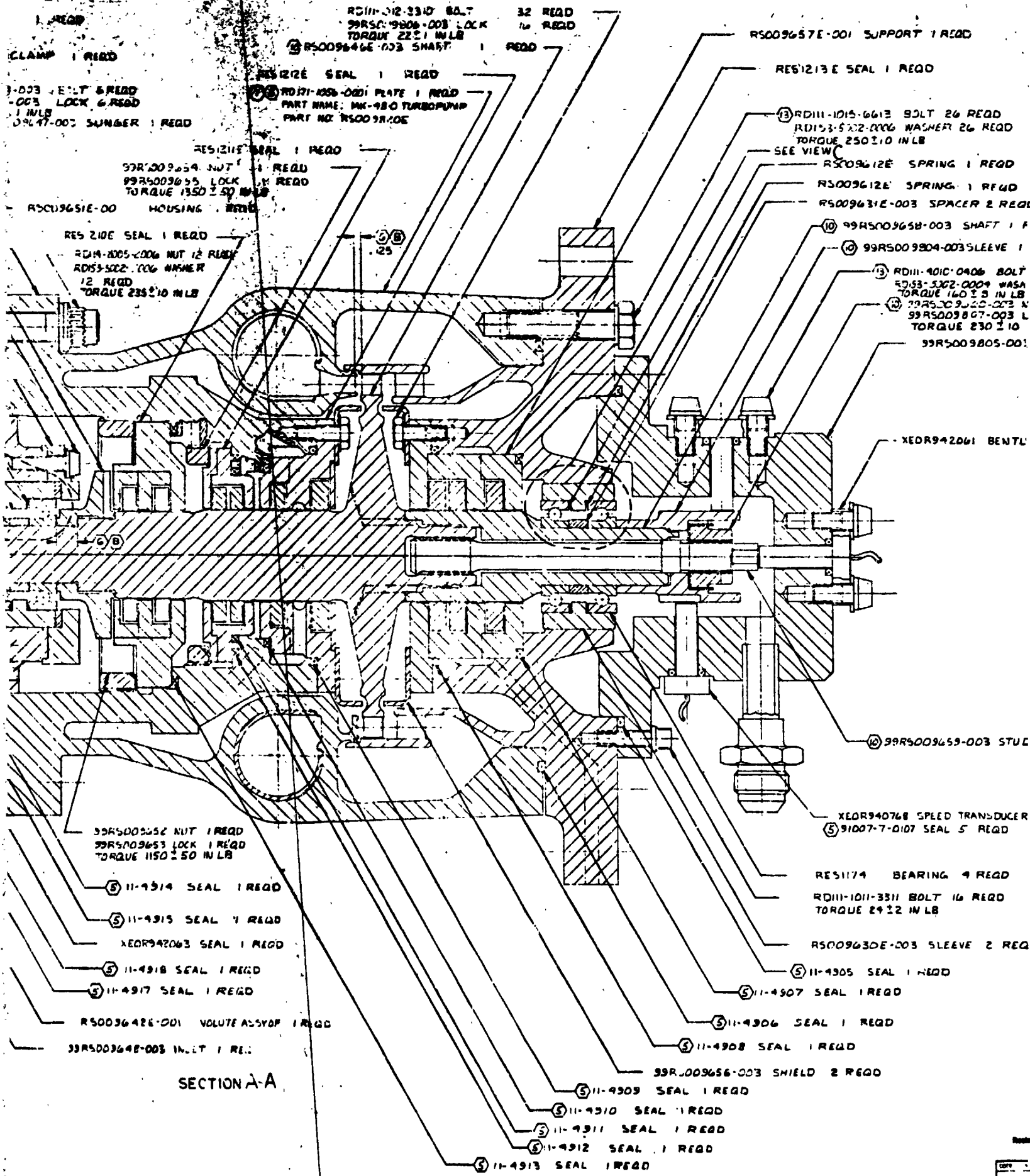
99R5009648-00

99R500
99R500
TORQUE
⑯ 24

Balluff International Corporation
Responsible Division
Sump-Pin Assembly

CODE IDENT OFFICE PARTS 1
R5009820E

EXHAUST FLARE



RD111-012-2310 BOLT 32 REQD
99R5009806-003 LOCK 16 REQD
TORQUE 2221 IN LB
99R5009846-003 SHAFT 1 REQD

RS009657E-001 SUPPORT 1 REQD

CLAMP 1 REQD
9-003 BOLT 6 REQD
003 LOCK 6 REQD
1 IN LB
09L47-000 SUMMER 1 REQD

RESIZIE SEAL 1 REQD

RESIZIE SEAL 1 REQD

RD171-006-0001 PLATE 1 REQD
PART NAME: MK-180 TURBOPUMP
PART NO: RS009820E

RD111-1015-6613 BOLT 26 REQD
RD153-502-0006 WASHER 26 REQD
TORQUE 250210 IN LB
SEE VIEW C

RESIZIE SEAL 1 REQD
99R5009854 NUT 1 REQD
99R5009855 LOCK 1 REQD
TORQUE 1350 250 IN LB

RS009612E SPRING 1 REQD

RS009651E-00 HOUSING 1 REQD

RS009612E SPRING 1 REQD

RS009631E-003 SPACER 2 REQD

RESIZIE SEAL 1 REQD
RD14-805-2006 NUT 12 REQD
RD153-502-0006 WASHER 12 REQD
TORQUE 235210 IN LB

99R5009858-003 SHAFT 1 REQD

99R5009804-003 SLEEVE 1 REQD

RD111-401C-0406 BOLT 20 REQD
RD153-502-0004 WASHER 20 REQD
TORQUE 160213 IN LB
99R5009807-003 LOCK 20 REQD
TORQUE 230210 IN LB
99R5009805-001

XEDR942061 BENTU

99R5009859-003 STUD

99R5009852 NUT 1 REQD
99R5009853 LOCK 1 REQD
TORQUE 1150 250 IN LB

XEDR940768 SPEED TRANSDUCER
91007-7-0107 SEAL 5 REQD

11-4914 SEAL 1 REQD

RES1174 BEARING 4 REQD

11-4915 SEAL 1 REQD

RD111-1011-3311 BOLT 16 REQD
TORQUE 2422 IN LB

XEDR942063 SEAL 1 REQD

RS009630E-003 SLEEVE 2 REQD

11-4918 SEAL 1 REQD

11-4905 SEAL 1 REQD

11-4917 SEAL 1 REQD

11-4907 SEAL 1 REQD

RS009642E-001 VOLUTE ASSY 1 REQD

11-4906 SEAL 1 REQD

99R5009848-003 INLET 1 REQD

11-4908 SEAL 1 REQD

SECTION A-A

99R5009856-003 SHIELD 2 REQD

11-4909 SEAL 1 REQD

11-4910 SEAL 1 REQD

11-4911 SEAL 1 REQD

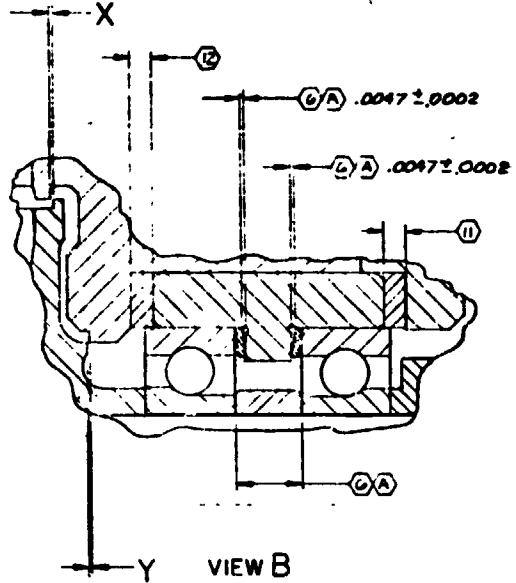
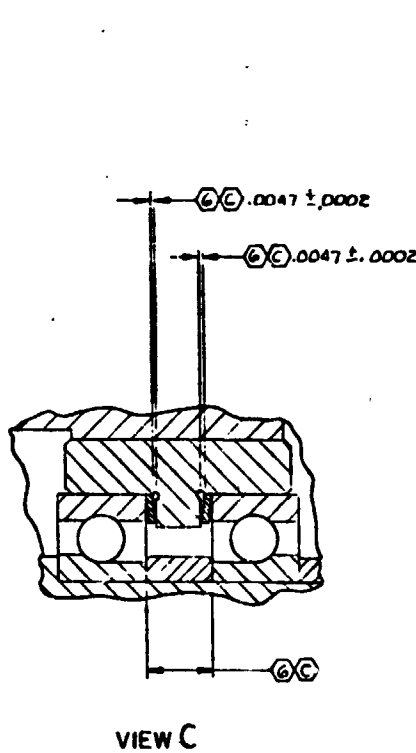
11-4912 SEAL 1 REQD

11-4913 SEAL 1 REQD

FOLDOUT FRAME 4

1 REQD

KER 4 REQD



- 13 LOCKWIRE BOLTS PER RADIO1-002.
- 12 MACHINE SPACER THICKNESS PER DETAIL DRAWING TO OBTAIN AT LIQUID NITROGEN TEMPERATURE A FORCE OF 420 ± 50 LB ON THE ROTOR WHEN CLEARANCE X IS ZERO.
- 11 MACHINE SPACER THICKNESS PER DETAIL DRAWING TO OBTAIN AT LIQUID NITROGEN TEMPERATURE A FORCE OF 470 ± 30 LB ON THE ROTOR WHEN CLEARANCE Y IS ZERO.
- 10 BALANCE ROTOR ASSEMBLY TO WITHIN .02 GRAV INCHES. MATCH MARK ALL ROTOR ASSEMBLY COMPONENTS PER RADIO4-008 BY ELECTRO CHEM ETCH. COATED PARTS ARE COMPONENTS OF THE ROTOR ASSEMBLY. SLIGN MATCHMARKS. INSTALL FITTINGS AND TUBE ASSEMBLIES PER RADIO2-003.
- 9 APPLY PER RADIO4-010.
- 7 IDENTIFY PER RADIO4-008.
- 6 MACHINE SPACER PER DETAIL DRAWING TO OBTAIN MATCH CODED DIMENSION.
- 5 HYDRODNE DIVISION, F.P.I. INC, NORW HOLLOWOOD, CAL.
- 4. INSTALL PROTECTIVE COVERS OVER ALL EXPOSED FLANGES, TUBE FITTINGS & PORTS.
- 3. LEAK CHECK TURBOPUMP PER DETAILED ASSEMBLY INSTRUCTIONS.
- 2. INSTALL THREADED FASTENERS PER RADIO1-002.
- 1. CLEAN ALL PARTS PRIOR TO PUMP ASSEMBLY PER RADIO1-004. ALL INTERNAL SURFACES TO BE MAINTAINED FOR CLEAN.

100% GRADE OVERLAP SPECIFIED

FOLDOUT FRAME 5

.0047 ± .0002
 A) .0047 ± .0002



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BOLTS PER RAD101-002.
 SPACER THICKNESS PER DETAIL DRAWING TO OBTAIN MATCH DIMENSION.
 PROTECTIVE COVERS OVER ALL EXPOSED FLANGES, FITTINGS & PORTS.
 CHECK TURBOPUMP PER DETAILED ASSEMBLY VIEWS.
 THREADED FASTENERS PER RAD101-001.
 ALL PARTS PRIOR TO PUMP ASSEMBLY PER ALL INTERNAL SURFACES TO BE MAINTAINED

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**MARK 48-0 Turbopump Assembly
 Drawing RS009820E**

105

NONE NONE	NONE	TURBOPUMP ASSY, OXIDIZER	
		J 02602	RS009820E

RS009820E

3006850E

MK48-0

APPENDIX C

TESTS NO. 004 AND
NO. 005 PRINTOUT DATA

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MK48
LIQUID OXYGEN TURBOPUMP ASSEMBLY

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COMMENTS . . .
ASSUMED 1ST AND 2ND SLICES OF PUMP DISCH PR (251 PSI)
ASSUMED TURB IN TEMP FOR ALL SLICES (510 R)
ASSUMED PRI HG SL CRF PR FOR SLICES 1,2,3,4 AND 5 (1.0 PSI)

AMBIENT PRESSURE 13.8000

LO2 VENTURI (GG)
P/N V160248-SGR
S/N 8871
UPSTREAM DIAMETER 0.9570
THROAT DIAMETER 0.2480
THROAT CD 0.9850

GH2 VENTURI (TURB)
P/N VPJ3120J-SGR
S/N 9731
UPSTREAM DIAMETER 2.3030
THROAT DIAMETER 1.3085
THROAT CD 0.9873

LH2 VENTURI (GG)
P/N V320471-SGR
S/N 8873
UPSTREAM DIAMETER 1.6890
THROAT DIAMETER 0.4710
THROAT CD 0.9765

LOX VENTURI (PUMP DISCH)
P/N V321J59-SGR
S/N 8877
UPSTREAM DIAMETER 1.6890
THROAT DIAMETER 1.0590
THROAT CD 0.9820

BAL PIST OVBD CONT ORIFICE 0.2090
BAL PIST OVBD ORIFICE 0.2210
LOX SEAL DRAIN ORIFICE 1.1263
TURBINE EFFECTIVE AREA 0.2319
TURBINE EXHAUST ORIFICE 0.6600
PRIMARY H.G. DRAIN ORIFICE 0.6550
SEC. H.G. DRAIN ORIFICE 0.6000
REAR BRG COOLANT ORIFICE 0.1800
INTERMEDIATE SEAL PURGE ORIFICE 0.1260

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G A S E O U S H Y D R O G E N T U R B I N E D R I V E P A R A M E T E R S

SLICE NO	BEGIN TIME (SEC)	END TIME (SEC)	VENTURI U/S PR (PSIA)	VENTURI U/S PR (DEG R)	VENTURI DFLTA PR (PSID)	SPIN VALVE POSN	SPIN VALVE U/S PR (PSIA)	SPIN VALVE DELTA P (PSID)	TURB GHZ FLOW (LB/SEC)	SPEED (RPM)
1	40.007	40.523	4071.1	530.2	0.00	-1.368	4059.1	4022.254	0.02941	2812.
2	48.009	48.524	4051.9	530.2	0.00	-1.375	4039.7	4005.342	0.04400	2814.
3	60.011	60.526	4014.0	530.3	0.07	0.634	4001.9	3883.544	0.27593	9102.
4	70.033	70.545	3974.1	530.5	0.26	3.026	3959.1	3659.074	0.54225	19548.
5	80.014	80.530	3914.6	530.5	0.43	4.175	3899.4	3492.464	0.69257	23523.
6	90.027	90.552	3835.0	530.5	0.88	6.286	3818.1	3201.234	0.98134	29751.
7	100.018	100.533	3748.9	530.2	0.85	6.312	3733.1	3127.385	0.95579	29379.
8	112.020	112.535	3650.6	529.8	0.83	6.365	3635.1	3043.966	0.93692	29143.
9	120.021	120.537	3587.7	529.6	0.84	6.466	3572.6	2977.612	0.93441	29176.
10	127.033	127.548	3523.2	529.3	0.85	6.548	3518.8	2925.020	0.93318	29345.
11	134.033	134.560	3480.7	529.0	0.87	6.678	3465.6	2867.325	0.93936	29746.
12	139.035	139.551	3443.3	528.8	0.88	6.692	3429.1	2835.672	0.93330	29775.
13	147.036	147.552	3385.3	528.5	0.88	6.796	3370.7	2776.772	0.92901	30473.
14	153.017	153.532	3342.3	528.2	0.88	6.863	3328.3	2735.652	0.92237	30388.
15	159.039	159.554	3299.7	527.9	0.89	6.983	3285.9	2689.717	0.92531	29318.
16	164.029	164.545	3264.5	527.6	0.89	7.018	3250.8	2658.645	0.91944	29238.
17	170.010	170.525	3222.7	527.2	0.89	7.094	3209.7	2617.571	0.91696	28741.
18	174.010	174.526	3195.0	527.0	0.90	7.141	3181.8	2589.869	0.91516	28723.

PK48
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T U R B I N E P A R A M E T E R S

SLICE NO	BEGIN TIME (SEC)	END TIME (SEC)	SPEED (RPM)	TURB INLET (PSIA)		TURB INLET (DEG R)		TURB INLET (PSIA)		TURB EXH (PSIA)		TURB EXH (DEG R)		TURB ORIFICE DELTA P (PSID)	
				TOT PR	STAT PR	TOT PR	STAT PR	TOT PR	STAT PR	TOT PR	STAT PR	TOT PR	STAT PR	TOT PR	STAT PR
1	40.007	40.523	2812.	35.6	43.0	510.0	37.7	39.1	33.6	448.9	19.9				
2	48.009	48.524	2814.	36.2	42.8	510.0	35.9	38.3	30.8	448.8	17.1				
3	60.011	60.526	9102.	120.4	135.1	510.0	101.8	106.3	95.6	451.4	81.8				
4	70.033	70.549	19548.	305.6	313.7	510.0	215.3	222.3	210.8	330.9	197.0				
5	80.014	80.530	23523.	406.8	418.5	510.0	280.6	286.6	275.4	393.7	261.7				
6	90.037	90.552	29751.	618.0	624.8	510.0	411.5	418.9	407.6	435.3	393.8				
7	100.018	100.533	29379.	604.6	613.2	510.0	405.7	410.2	400.4	435.6	386.7				
8	112.020	112.535	29143.	594.9	602.6	510.0	399.4	405.5	393.9	435.0	380.3				
9	120.021	120.537	29176.	556.3	602.6	510.0	359.3	405.4	393.9	435.1	380.3				
10	127.033	127.548	29345.	594.4	632.5	510.0	398.8	435.2	393.6	454.6	380.0				
11	134.003	134.560	29746.	555.7	604.2	510.0	400.0	405.8	394.7	434.4	381.0				
12	139.035	139.551	29775.	592.1	602.4	510.0	396.2	402.2	393.5	433.5	379.8				
13	147.036	147.552	30473.	595.3	632.4	510.0	396.0	431.9	393.5	433.0	379.9				
14	153.017	153.532	30388.	592.1	601.4	510.0	395.4	401.2	392.7	432.6	379.1				
15	159.039	159.554	29318.	593.6	603.4	510.0	398.9	404.7	394.0	433.2	380.3				
16	164.029	164.545	29238.	594.1	601.3	510.0	395.7	401.3	392.9	433.0	379.3				
17	170.010	170.525	28741.	591.0	600.8	510.0	395.9	401.3	393.2	432.8	379.6				
18	174.010	174.526	28723.	590.0	600.1	510.0	395.2	400.8	392.2	433.3	378.7				

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 TEST DATE 05-23-78

TURBINE PARAMETERS (CONTINUED)

TIME SLICE NO	SPEED RPM	FLOW (LB/SEC)	PR RATIO (T-S)	BHP (CALIB) (HP)	BHP (DELTA T) (HP)	TORQUE (FT-LB)	U/C (T-S)	EFF (T-S)	TURB NJZ IN TOT P (PSIA)
1	2812.	0.0294	1.0765	0.2	9.1	0.31	0.0425	10.88	42.1
2	2814.	0.0440	1.0953	0.3	13.6	0.51	0.0382	9.70	41.9
3	9102.	0.2799	1.2456	8.5	83.3	4.91	0.0804	19.95	132.4
4	19548.	0.5422	1.3828	37.4	493.9	10.05	0.1428	30.98	307.4
5	23523.	0.6926	1.4311	58.0	410.1	12.95	0.1635	34.05	410.1
6	29751.	0.9813	1.4616	100.1	374.4	17.67	0.2006	39.05	612.3
7	29379.	0.9558	1.4650	97.1	363.1	17.35	0.1976	38.67	601.0
8	29163.	0.9369	1.4561	93.7	358.5	16.88	0.1975	38.66	590.5
9	29176.	0.9344	1.4568	93.6	357.5	16.84	0.1976	38.67	590.6
10	29345.	0.9332	1.4570	93.8	359.2	16.79	0.1987	38.81	590.4
11	29746.	0.9394	1.4593	95.5	362.4	16.86	0.2011	39.11	592.1
12	25775.	0.9353	1.4678	96.1	365.5	16.95	0.1998	38.95	590.3
13	30473.	0.9290	1.4690	91.0	365.1	16.72	0.2043	39.51	590.3
14	30388.	0.9234	1.4693	96.3	364.7	16.64	0.2037	39.43	589.4
15	29318.	0.9253	1.4611	93.4	362.8	16.73	0.1979	38.70	591.4
16	29238.	0.9194	1.4687	93.5	361.5	16.78	0.1961	38.47	589.3
17	28741.	0.9170	1.4671	92.0	361.6	16.81	0.1930	38.07	588.2
18	28723.	0.9152	1.4674	91.8	358.2	16.78	0.1928	38.05	588.1

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T U R B I N E P A R A M E T E R S (CONTINUED)

TIME SLICE NO	SPEED RPM	AVAIL. ENERGY (BTU/LB)	GAMMA	CP (BTU/LBM-R)	SPEED PARA-METER	FLOW PARA-METER	TORQUE PARA-METER	TURB WHEEL EXH PRESS PSIA
1	2812.	36.75	1.3808	3.5795	21.1	1.8836	0.0068	42.8
2	2814.	45.24	1.3808	3.5795	21.1	2.8288	0.0112	41.5
3	9102.	107.62	1.3817	3.5843	68.4	5.7309	0.0340	110.7
4	19548.	157.42	1.3834	3.5934	146.9	4.7570	0.0300	228.5
5	23523.	173.83	1.3843	3.5986	176.8	4.5542	0.0290	293.3
6	29751.	184.61	1.3861	3.6085	223.6	4.3225	0.0265	427.4
7	29379.	185.64	1.3860	3.6080	220.8	4.2894	0.0265	419.0
8	29143.	182.78	1.3859	3.6075	219.0	4.2791	0.0262	413.8
9	29176.	183.01	1.3859	3.6075	219.3	4.2672	0.0262	413.4
10	29343.	183.07	1.3859	3.6075	220.5	4.2627	0.0261	413.0
11	29746.	183.79	1.3859	3.6076	223.6	4.2786	0.0261	414.0
12	29775.	186.45	1.3859	3.6075	223.8	4.2731	0.0263	410.7
13	30473.	186.83	1.3859	3.6075	229.0	4.2444	0.0260	410.5
14	30388.	186.93	1.3859	3.6074	228.4	4.2251	0.0259	409.7
15	29318.	184.37	1.3859	3.6075	220.3	4.2233	0.0260	413.2
16	29238.	186.72	1.3859	3.6074	219.7	4.2078	0.0261	410.1
17	28741.	186.25	1.3859	3.6074	216.0	4.2003	0.0262	409.8
18	28723.	186.33	1.3859	3.6074	215.9	4.1969	0.0262	409.2

MK46
LIQUID OXYGEN TURBOPUMP ASSEMBLY

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SEAL AND BEARING DATA (CONTINUED)

TIME SLICE NO	SEAL DR FLOW (LB/SEC)	RR BPG COOLANT SUPPL. PR (PSIA)	RR BRG COOLANT SUPPL. TEMP (DEG R)	RR DRG PR (PSIA)	RR BRG DRAIN TEMP (DEG R)	RR BRG COOL. DRIF U/S PR (PSIA)	RR BRG COOL. DRIF U/S TEMP (DEG R)	RR BRG COOLANT FLOW (LB/SEC)
1	0.0	3540.1	141.04	215.4	107.84	13.8	460.00	0.0
2	0.0	3534.7	141.65	208.3	99.16	13.8	460.00	0.0
3	0.0	3533.2	143.21	201.2	90.93	13.8	460.00	0.0
4	0.0	3546.1	143.60	205.0	88.26	13.8	460.00	0.0
5	0.0	3553.6	143.73	212.7	89.64	13.8	460.00	0.0
6	0.0	3557.5	144.30	214.8	89.57	13.8	460.00	0.0
7	0.0	3557.1	144.40	213.3	8.39	13.8	460.00	0.0
8	0.0	3556.3	145.12	212.6	37.80	13.8	460.00	0.0
9	0.0	3522.7	145.86	211.5	88.03	13.8	460.00	0.0
10	0.0	3469.9	146.64	208.1	86.74	13.8	460.00	0.0
11	0.0	3410.4	147.50	205.6	86.08	13.8	460.00	0.0
12	0.0	311.9	148.20	202.6	84.24	13.8	460.00	0.0
13	0.0	3324.2	149.64	199.9	83.60	13.8	460.00	0.0
14	0.0	3282.5	150.93	197.2	82.64	13.8	460.00	0.0
15	0.0	3240.6	152.38	194.6	81.47	13.8	460.00	0.0
16	0.0	3206.1	153.79	193.7	81.56	13.8	460.00	0.0
17	0.0	3165.5	155.65	189.7	79.54	13.8	460.00	0.0
18	0.0	3137.7	156.93	190.2	80.78	13.8	460.00	0.0

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SEAL AND BEARING DATA (CONTINUED)

TIME SLICE NO	LOX SEAL DRAIN ORIF U/S PRESS PSIA	LOX SEAL DRAIN ORIF U/S TEMP DEG R	LOX SEAL DRAIN FLOWRATE (LB/SEC)
1	11.0	506.58	0.0
2	11.0	506.19	0.0
3	11.0	508.97	0.0
4	10.9	512.10	0.0
5	10.9	514.13	0.0
6	10.9	514.93	0.0
7	10.9	515.85	0.0
8	10.9	516.89	0.0
9	10.9	517.41	0.0
10	10.9	517.91	0.0
11	10.9	518.32	0.0
12	10.9	518.59	0.0
13	10.9	519.05	0.0
14	10.9	519.45	0.0
15	10.9	519.81	0.0
16	10.9	520.02	0.0
17	10.9	520.27	0.0
18	10.9	520.53	0.0

MK66
LIQUID OXYGEN TURBOPUMP ASSEMBLY

PROCESSING DATE 05-31-78
TEST CURATION, SEC 124.00

RUN NUMBER 4
TEST DATE 05-23-78

PUMP PRESSURES

TIME SLICE NO	SUPPLY TANK PR (PSIA)	PUMP INFLT PR (PSIA)	IMPELLER DISCH PR (PSIA)	DIFFUSER DISCH PR (PSIA)	PUMP DISCH PR (PSIA)	BAL PIST CAV PR #1 (PSIA)	BAL PIST SUMP PR (PSIA)	VAL PIST RETURN PR (PSIA)
1	255.9	254.9	168.8	177.2	264.8	227.3	100.7	79.4
2	256.2	255.3	168.0	177.3	264.8	224.0	97.3	75.6
3	255.9	252.3	211.4	253.0	272.9	283.7	117.1	87.2
4	255.1	246.9	386.6	519.7	570.0	494.9	185.0	150.7
5	253.9	241.7	485.9	656.4	728.3	604.3	245.6	208.2
6	253.9	234.1	653.1	900.7	1026.5	800.2	351.2	305.6
7	254.1	233.7	640.3	828.9	983.6	764.2	347.7	303.1
8	254.2	232.9	605.8	793.6	968.6	739.0	348.4	303.9
9	254.1	234.0	608.1	796.6	978.9	730.5	350.7	305.7
10	254.1	235.1	621.6	821.7	1007.4	730.4	357.3	311.2
11	254.1	237.0	648.5	876.7	1073.9	751.5	370.3	321.3
12	254.0	237.5	645.5	881.4	1084.1	748.3	370.4	323.5
13	254.0	240.7	683.6	968.5	1189.1	780.3	387.3	337.2
14	253.9	240.2	685.8	949.5	1175.4	767.2	385.8	335.9
15	253.5	233.7	612.1	796.0	994.5	686.0	358.3	311.2
16	253.8	224.0	618.2	793.8	989.3	678.5	357.0	310.8
17	254.0	230.8	571.2	703.9	886.8	636.0	337.1	294.0
18	254.1	230.4	569.8	700.8	882.4	632.1	336.8	293.3

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MK41
 LIQUID OXYGEN TURBOPUMP ASSEMBLY

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 PROCESSING DATE 05-31-73
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PUMP PRESSURES AND TEMPERATURES

TIME SLICE NO	SPEED RPM	INDUCER		PUMP BRG		BAL PIST	
		DISCH PRESS PSIA	PRESS PSIA	D/S PRESS PSIA	RECIRC TEMP DEG R	RECIRC TEMP DEG R	RECIRC TEMP DEG R
1	2812.	223.8	293.4	102.9	291.7	205.56	
2	2814.	224.1	291.4	100.4	287.7	205.51	
3	9102.	236.1	295.0	108.6	301.3	201.58	
4	19548.	269.9	303.9	155.2	301.2	203.56	
5	23523.	281.6	324.4	202.4	312.3	207.02	
6	29751.	304.3	329.0	287.4	316.6	213.07	
7	29379.	293.4	361.2	291.7	329.3	209.37	
8	29143.	291.4	357.2	287.7	328.6	212.58	
9	29176.	295.0	298.5	301.3	304.1	212.89	
10	25345.	303.9	297.4	301.2	303.6	208.34	
11	25746.	324.4	262.8	312.3	287.8	208.30	
12	29775.	329.0	260.9	316.6	287.8	211.63	
13	30473.	361.2		329.3	218.56	218.56	
14	30388.	357.2		328.6	220.59	220.59	
15	29218.	298.5		304.1	221.23	221.23	
16	29238.	297.4		303.6	214.43	214.43	
17	28741.	262.8		287.8	219.38	219.38	
18	28723.	260.9		287.8	212.68	212.68	

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MK46
LIQUID OXYGEN TURBOPUMP ASSEMBLY

PROCESSING DATE 05-31-76
TEST DURATION, SEC 124.00

RUN NUMBER 4
TEST DATE 05-23-78

PUMP PRESSURES AND TEMPERATURES (CONTINUED)

TIME SLICE NO	BAL PIST		BAL PIST		BAL PIST		BAL PIST FLCM	BAL PIST LB/SEC
	OVBD PRFSS PSIA	TEMP DEG R	OVBD DELTA P PSI	OVBD DELTA P PSI	OVBD FLCM	OVBD LB/SEC		
1	90.4	174.4	17.20				0.871	
2	87.2	174.4	16.48				0.852	
3	95.3	175.4	18.33				0.898	
4	147.0	182.3	31.15				1.160	
5	194.5	186.9	42.95				1.354	
6	277.0	196.2	62.59				1.614	
7	274.7	195.7	61.68				1.603	
8	274.2	195.1	61.54				1.603	
9	276.6	195.3	61.59				1.617	
10	280.5	195.5	62.70				1.643	
11	290.0	196.2	64.86				1.647	
12	290.8	196.3	65.18				1.680	
13	322.7	197.8	68.12				1.673	
14	301.8	197.7	67.57				1.611	
15	280.4	195.7	62.28				1.508	
16	279.9	195.5	61.98				1.566	
17	266.3	194.9	58.75				1.565	
18	265.9	194.9	58.65					

MK46
LIQUID OXYGEN TURBOPUMP ASSEMBLY

RUN NUMBER 4
TEST DATE 05-23-78
PROCESSING DATE 05-31-78
TEST DURATION, SEC 124.00

PUMP PRESSURES AND TEMPERATURES

TIME SLICE NO	PUMP INLET TEMP (DEG R)		PUMP DISCH TEMP (DEG R)		BAL PIST RETURN TEMP (DEG R)		PUMP DISCH VENT U/S TEMP (DEG R)		PUMP DISCH VENT U/S PR (PSIA)		PUMP DISCH VENT DELTA PR (PSI)		HP (TEMP)
	(DEG R)	(DEG R)	(DEG R)	(DEG R)	(DEG R)	(DEG R)	(DEG R)	(DEG R)	(PSIA)	(PSIA)	(PSI)	(PSI)	
1	172.37	172.75	172.75	173.56	181.01	173.56	193.2	193.2	0.82	0.82	1.35		
2	172.40	172.78	173.69	173.69	181.21	173.69	193.6	193.6	0.66	0.66	1.21		
3	172.45	173.06	173.74	173.74	178.59	173.74	276.8	276.8	1.30	1.30	2.68		
4	172.29	174.18	174.54	174.54	182.03	174.54	571.5	571.5	2.57	2.57	16.97		
5	172.07	174.74	175.12	175.12	186.18	175.12	726.3	726.3	4.70	4.70	32.55		
6	171.97	176.24	176.50	176.50	194.47	176.50	1020.0	1020.0	7.60	7.60	66.33		
7	171.97	176.14	176.49	176.49	194.15	176.49	977.7	977.7	8.26	8.26	66.33		
8	172.00	176.14	176.47	176.47	193.83	176.47	962.1	962.1	8.08	8.08	64.81		
9	172.07	176.18	176.56	176.56	194.08	176.56	972.6	972.6	7.87	7.87	64.10		
10	172.07	176.23	176.62	176.62	194.38	176.62	1001.2	1001.2	7.37	7.37	63.77		
11	172.07	176.43	176.75	176.75	194.81	176.75	1059.1	1059.1	6.35	6.35	63.48		
12	172.07	176.47	176.78	176.78	195.13	176.78	1079.1	1079.1	6.18	6.18	63.35		
13	172.13	176.83	177.17	177.17	196.47	177.17	1186.3	1186.3	4.62	4.62	60.96		
14	172.18	176.82	177.22	177.22	196.47	177.22	1171.9	1171.9	4.78	4.78	60.97		
15	172.12	176.33	176.69	176.69	194.80	176.69	987.8	987.8	8.16	8.16	66.76		
16	172.12	176.33	176.66	176.66	194.49	176.66	982.1	982.1	7.72	7.72	64.80		
17	172.14	176.16	176.51	176.51	194.10	176.51	877.9	877.9	9.62	9.62	65.73		
18	172.17	176.17	176.50	176.50	194.09	176.50	874.1	874.1	9.42	9.42	64.78		

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MK48
LIQUID OXYGEN TURBOPUMP ASSEMBLY

PROFESSING DATE J5-31-78
TEST DURATION, SEC 124.00

RUN NUMBER 4
TEST DATE 05-23-78

C A L C U L A T E D P U M P P A R A M E T E R S

TIME SLICE NO	TEST SPEED (RPM)	PUMP FLOW (GPM)	PUMP FLOW (LB/SEC)	HEAD FT	PRESS RISE (PSI)	FLUID HP (MP)	TURB HP (MP)	EFF (%)	ISEN EFF (%)	SPECIFIC SPEED
1	2812.	30.42	4.7057	22.1	9.9	0.2	0.17	113.83	14.02	1520.
2	2814.	27.29	4.2203	21.0	9.5	0.2	0.27	59.05	13.38	1497.
3	9102.	38.21	5.9172	43.4	19.6	0.5	8.51	5.49	17.37	3330.
4	19548.	53.59	8.3188	672.5	323.1	10.2	37.42	27.19	59.94	1084.
5	23523.	72.46	11.2593	1014.1	486.7	20.8	58.01	35.80	63.79	1114.
6	29751.	92.10	14.3242	1650.2	792.4	43.0	100.12	42.94	64.79	1103.
7	29379.	96.03	14.9267	1564.2	749.9	42.5	97.08	43.75	64.00	1157.
8	29143.	95.02	14.7670	1532.8	734.7	41.2	93.68	43.95	63.50	1160.
9	29176.	93.76	14.5699	1553.5	744.9	41.2	93.57	44.00	64.20	1142.
10	29345.	90.74	14.1046	1608.7	772.3	41.3	93.82	43.99	64.69	1100.
11	25746.	84.19	13.0555	1739.4	836.9	41.4	95.53	43.37	65.24	1013.
12	29775.	83.37	12.9224	1758.9	846.5	41.3	96.11	42.01	65.24	999.
13	30472.	71.82	11.1794	1965.1	948.4	39.9	97.04	41.18	65.53	875.
14	30388.	73.03	11.3644	1938.5	935.2	40.1	96.31	41.60	65.70	889.
15	29318.	55.49	14.8385	1587.1	760.8	42.8	93.42	45.85	64.14	1139.
16	29238.	92.88	14.4317	1574.3	755.0	41.3	93.46	44.22	63.75	1127.
17	28141.	103.75	16.1018	1374.6	655.9	40.2	92.01	43.76	61.22	1297.
18	28723.	102.66	15.9327	1366.1	651.9	39.6	91.81	43.12	61.09	1295.

MK46
LIQUID OXYGEN TURBOPUMP ASSEMBLY

PROCESSING DATE 05-31-78
TEST DURATION, SEC 124.00

RUN NUMBER 4
TEST DATE 05-23-78

CALCULATED PUMP PARAMETERS

TIME SLICE NO	TEST SPEED (RPM)	NPSH (FT)	SUCTION SPECIFIC SPEED	INLET FLOW COEFF	HEAD COEFF	PUMP DELTA T (DEG R)	(O/N) OVER (O/N)DES
1	2812.	474.16	152.63	0.27491	0.72720	0.38	3.2634
2	2814.	474.95	144.48	0.24647	0.69009	0.37	2.9258
3	9102.	470.94	556.58	0.10671	0.13599	0.61	1.2667
4	19548.	457.82	1445.79	0.36968	0.45739	1.91	3.8271
5	23523.	447.12	2059.35	0.07830	0.47629	2.67	0.9295
6	29751.	431.81	3014.14	0.37868	0.48452	4.28	0.9340
7	29379.	431.16	3042.71	0.08338	0.47399	4.18	0.9862
8	29142.	431.56	3000.25	0.08287	0.46903	4.14	0.9838
9	29176.	431.83	2982.25	0.08168	0.47429	4.11	0.9696
10	29345.	423.93	2940.05	0.07859	0.48551	4.17	0.9330
11	29746.	437.73	2851.93	0.07193	0.51091	4.36	0.8539
12	29775.	438.78	2830.61	0.07091	0.51561	4.40	0.8418
13	30473.	445.19	2664.48	0.05900	0.54999	4.70	3.7111
14	30388.	444.22	2683.82	0.06108	0.54556	4.64	0.7251
15	29318.	431.24	3027.42	0.08279	0.47988	4.22	0.5828
16	29238.	431.77	2974.84	0.08074	0.47861	4.71	0.9584
17	28741.	425.61	3124.15	0.09175	0.43250	4.02	1.0892
18	28723.	424.79	3110.25	0.09085	0.43035	4.00	1.0785

MK45
LIQUID OXYGEN TURBOPUMP ASSEMBLY

PROCESSING DATE 05-31-78
TEST DURATION, SEC 124.00

RUN NUMBER 4
TEST DATE 05-23-78

SCALED TO TARGET SPEED = 3000. RPM

TIME SLICE NO	FLOW (GPM)	FLOW (M/SEC)	HEAD (FT)	PRESS RISE (PSI)	HORSE POWER (BHP)	NPSH (FT)
1	324.47	50.19926	2518.28	1123.44	1639.32	53961.26
2	290.91	44.95289	2389.78	1079.86	1460.89	53581.09
3	125.55	19.50264	470.93	213.45	96.11	5115.82
4	82.24	12.76697	1583.92	760.91	61.34	1078.32
5	92.42	14.35565	1649.40	791.58	67.51	727.26
6	92.97	14.44386	1677.90	805.68	68.01	439.05
7	98.16	15.24222	1631.05	781.91	70.63	449.58
8	97.81	15.20139	1624.26	778.55	70.70	457.32
9	96.40	14.98133	1642.48	787.59	69.69	456.56
10	92.76	14.41953	1681.32	837.15	68.14	453.53
11	84.90	13.20740	1769.28	851.28	65.13	445.24
12	83.70	13.02009	1785.55	859.39	64.79	445.44
13	70.70	11.06586	1904.62	919.19	58.16	431.48
14	72.10	11.21927	1889.27	911.43	58.66	432.94
15	97.71	15.18375	1661.81	796.61	71.53	451.55
16	95.30	14.80761	1657.43	794.89	73.70	454.56
17	108.30	16.80744	1497.76	714.68	74.76	463.73
18	107.23	16.64136	1490.31	711.23	73.82	463.42

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MK44

LIQUID OXYGEN TURBOPUMP ASSEMBLY

PROCESSING DATE 05-31-78
TEST DURATION, SEC 124.00

RUN NUMBER 4
TEST DATE 05-23-78

SCALED TO DESIGN SPEED = 70,000. RPM

TIME SLICE NO	FLOW (GPM)	FLOW (M/SEC)	HEAD (FT)	PRESS RISE (PSI)	HORSE POWER (BHP)	NPSH (FT)
1	757.10	117.13161	13710.64	6116.48	20825.38	293789.10
2	678.79	104.98340	13011.04	5879.26	18558.73	293897.05
3	293.88	45.50617	2563.95	1162.14	1220.55	27852.80
4	191.90	29.78960	8623.59	4142.71	779.24	5873.85
5	215.64	33.50585	8980.07	4309.71	857.66	3959.52
6	216.69	33.70234	9135.24	4386.47	863.98	2350.39
7	228.81	35.56518	8880.15	4257.77	897.24	2447.72
8	228.23	35.46991	8843.18	4238.79	878.10	2489.84
9	224.94	34.95643	8942.37	4287.90	885.25	2485.71
10	216.45	33.64551	9153.83	4394.51	865.60	2469.21
11	198.11	30.81726	9632.77	4634.77	827.33	2424.11
12	195.29	30.38020	9721.35	4678.88	823.13	2425.17
13	164.97	25.68033	10369.58	5034.51	738.90	2349.18
14	168.22	26.17830	10286.02	4962.23	745.21	2357.13
15	228.00	35.42895	9047.63	4337.11	908.64	2458.43
16	222.36	34.55108	9029.79	4327.71	889.24	2474.82
17	252.69	38.21735	8154.48	3891.04	949.73	2524.73
18	250.20	38.82583	8113.91	3872.23	937.75	2523.07

MK66
LIQUID OXYGEN TURBOPUMP ASSEMBLY

RUN NUMBER 6
TEST DATE 05-31-78
PROCESSING DATE 06-01-78
TEST DURATION, SEC 43.00

COMMENTS . . .

REDUCED SLICES 1,2,3,4,5,6,7,10,17,25,42,50,52,54, AND 56.
PI0 63(PUMP DSCH VEN DPI) WAS CALCULATED AS FOLLOWS
159.1*(1/BECKMAN-133.951/13032.03-133.951)

AMBIENT PRESSURE

13.8000

LO2 VENTURI (GG)
P/N V160248-SGR
S/N 9871

UPSTREAM DIAMETER 0.9570
THROAT DIAMETER 0.2480
THROAT CD 0.9850

GH2 VENTURI (TURB)
P/N VP031200-SGR
S/N 9731

UPSTREAM DIAMETER 2.3000
THROAT DIAMETER 1.3085
THROAT CD 0.5873

LH2 VENTURI (GG)
P/N V320471-SGR
S/N 8873

UPSTREAM DIAMETER 1.6890
THROAT DIAMETER 0.4710
THROAT CD 0.9765

LOX VENTURI (PUMP DISCH)
P/N V321059-SGR
S/N 8877

UPSTREAM DIAMETER 1.6890
THROAT DIAMETER 1.0590
THROAT CD 0.9820

BAL PIST OVRD CONT ORIFICE 0.2090
BAL PIST OVRD ORIFICE 0.2210
LOX SEAL DRAIN ORIFICE 1.1263
TURBINE EFFECTIVE AREA 0.2319
TURBINE EXHAUST ORIFICE 0.6600
PRIMARY H.G. DRAIN ORIFICE 0.6550
SEC. H.G. DRAIN ORIFICE 0.6000
REAR BRG COULANT ORIFICE 0.1810
INTERMEDIATE SEAL PURGE ORIFICE 0.1260

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MK44
LIQUID OXYGEN TURBOPUMP ASSEMBLY

RUN NUMBER 6
TEST DATE 05-31-78

PROCESSING DATE 06-01-78
TEST DURATION, SEC 43.00

G A S E Ū S H Y D R C E N T U R B I N E D R I V E P A R A M E T E R S

TIME SLICE NO	BEGIN TIME (SEC)	END TIME (SEC)	VENTURI U/S PR (PSIA)	VENTURI U/S TEMP (DEG R)	VENTURI DELTA PR (PSID)	SPIN VALVE F.U.N	SPIN VALVE U/S PR (PSIA)	SPIN VALVE DELTA P (PSID)	TURB GH2 FLOW (LB/SEC)	SPEED (RPM)
1	40.007	43.523	4224.0	542.4	0.83	5.652	429.4	3669.692	0.95059	29509.
2	45.039	45.554	4176.9	542.7	0.82	5.666	4243.5	3629.830	0.93712	29568.
3	50.029	50.545	4130.7	542.8	0.83	5.747	4197.2	3576.274	0.94183	29689.
4	55.023	55.536	4084.0	542.9	0.85	5.857	4150.4	3528.288	0.94366	29748.
5	60.011	60.526	3980.3	543.6	15.62	31.549	4008.8	875.350	4.00364	64519.
6	60.547	60.567	3969.0	543.6	15.86	32.075	3996.9	844.468	4.02843	65035.
7	60.588	63.639	3968.3	543.6	15.86	32.152	3996.2	842.034	4.02813	65176.
8	60.712	60.732	3963.3	543.6	15.88	32.306	3990.5	833.803	4.02938	65141.
9	61.000	61.021	3951.9	543.6	16.04	32.737	3980.0	810.269	4.04396	65504.
10	61.333	61.351	3940.4	543.4	16.16	33.110	3968.7	792.942	4.05470	65657.
11	62.032	62.052	3914.7	543.2	16.48	33.883	3940.5	753.512	4.08441	65823.
12	62.361	62.382	3900.4	543.1	16.56	34.239	3928.5	738.936	4.08789	66032.
13	62.444	62.547	3897.1	543.1	16.46	34.280	3925.7	732.948	4.07425	66016.
14	64.011	64.527	3831.7	542.2	16.76	35.776	3858.9	661.253	4.08511	66032.
15	65.579	66.053	3774.7	541.4	16.96	37.146	3802.5	604.787	4.08587	63163.

MK4E
LIQUID OXYGEN TURBOPUMP ASSEMBLY

RUN NUMBER 6
TEST DATE 05-31-78
PROCESSING DATE 06-01-78
TEST DURATION, SEC 43.00

TURBINE PARAMETERS

SLICE NO	BEGIN TIME (SEC)	END TIME (SEC)	SPEED (RPM)	TURB INLET (PSIA)		TURB INLET (DEG R)		TURB NOZZ (PSIA)		TURB EXH (PSIA)		TURB EXH (DEG R)		TURB DRIFTC (PSIA)	
				INLET (PSIA)	TOT PR (PSIA)	INLET (DEG R)	TOT PR (DEG R)	D/AS PR (PSIA)	STAT PR (PSIA)	EXH (PSIA)	STAT PR (PSIA)	TOT PR (PSIA)	EXH (DEG R)	TOT PR (DEG R)	DELTA P (PSIA)
1	40.007	40.523	29909.	619.4	626.8	490.2	413.7	423.3	409.3	440.1	442.8	393.2			
2	45.039	45.554	25568.	615.1	620.3	490.7	409.6	418.6	407.3	442.8	442.8	393.3			
3	53.029	53.545	29689.	617.7	626.7	490.8	414.1	423.3	410.0	444.4	444.4	396.0			
4	55.020	55.536	29748.	621.6	629.6	491.5	415.4	424.0	411.5	445.6	445.6	397.9			
5	60.011	60.526	64515.	3105.0	3090.3	501.4	1928.2	1941.9	1942.3	482.8	482.8	1903.2			
6	63.547	60.567	65335	3121.5	3108.7	501.6	1936.6	1950.1	1950.7	482.9	482.9	1911.2			
7	60.588	60.609	65176.	3124.4	3108.7	501.6	1934.5	1949.4	1950.7	482.8	482.8	1911.2			
8	60.712	60.732	65141.	3125.8	3110.2	501.6	1935.9	1950.8	1950.7	482.6	482.6	1911.2			
9	61.020	61.041	65534.	3138.4	3124.8	501.7	1942.9	1959.3	1958.4	482.1	482.1	1918.9			
10	61.330	61.351	65657.	3151.0	3129.7	502.0	1942.2	1958.6	1957.7	482.0	482.0	1918.1			
11	62.032	62.052	65833.	3159.4	3143.0	502.3	1949.9	1965.7	1965.5	481.9	481.9	1925.8			
12	62.361	62.382	66332.	3156.6	3145.1	502.4	1948.5	1966.4	1966.2	481.4	481.4	1926.4			
13	62.444	62.464	66016.	3162.2	3146.3	502.5	1951.0	1967.8	1968.0	481.5	481.5	1928.3			
14	64.011	64.031	66032.	3169.7	3150.0	503.6	1950.6	1968.2	1968.3	480.7	480.7	1928.4			
15	65.579	66.053	66163.	3167.6	3153.5	504.6	1953.5	1968.2	1968.3	479.8	479.8	1928.4			

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MK4
 LIQUID OXYGEN TURBOPUMP ASSEMBLY

 R/JN NUMBER 6
 TEST DATE 05-31-78
 PROCESSING DATE 06-01-78
 TEST DURATION, SEC 43.00

T U R B I N E P A R A M E T E R S (CONTINUED)

TIME SLICE NO	SPEED RPM	FLOW (LB/SEC)	PR RATIO (T-S)	BHP (CALIB) (HP)	BHP (DELTA T) (HP)	TORQUE (FT-LB)	U/C (T-S)	EFF (T-S)	TURB NOZ IN TOT P (PSIA)
1	29939.	0.9510	1.4512	93.8	245.4	16.47	0.2073	39.88	614.2
2	29568.	0.9371	1.4523	91.9	230.6	16.32	0.2047	39.56	607.9
3	29685.	0.9418	1.4510	92.5	224.4	16.36	0.2057	39.69	614.1
4	29748.	0.9437	1.4552	93.4	222.7	16.48	0.2052	39.63	617.0
5	64519.	4.0036	1.5595	645.1	389.9	52.49	0.3976	51.50	3028.5
6	65035.	4.0284	1.5622	651.2	394.4	52.57	0.3999	51.46	3046.6
7	65176.	4.0281	1.5628	651.5	396.8	52.48	0.4007	51.44	3046.6
8	65141.	4.0254	1.5624	651.4	401.9	52.53	0.4000	51.44	3047.9
9	65504.	4.0440	1.5630	653.9	415.7	52.41	0.4026	51.40	3062.3
10	65657.	4.0547	1.5660	658.7	426.6	52.67	0.4026	51.40	3067.1
11	65832.	4.0844	1.570	664.6	437.2	53.00	0.4033	51.39	3080.2
12	66032.	4.0879	1.5675	665.4	450.1	52.90	0.4043	51.36	3082.2
13	66016.	4.0743	1.5669	662.8	449.3	52.71	0.4043	51.36	3083.4
14	66032.	4.0851	1.5684	667.4	492.6	53.06	0.4036	51.38	3087.0
15	66163.	4.0859	1.5687	668.7	529.8	53.06	0.4041	51.37	3087.5

MK4
LIQUID OXYGEN TURBOPUMP ASSEMBLY

PROCESSING DATE 06-01-78
TEST DURATION, SEC 43.00

RUN NUMBER 6
TEST DATE 05-31-78

T U R B I N E P A R A M E T E R S (CONTINUED)

TIME SLICE NO	SPEED RPM	AVAIL. ENERGY(T-S) (BTU/LB)	GAMMA	CP (BTU/LBM-R)	SPEED PARA-METER	FLOW PARA-METER	TORQUE PARA-METER	TURB WHEEL EXH PRESS PSIA
1	29909.	174.79	1.3835	3.6353	229.3	4.0939	0.0246	428.1
2	29568.	175.24	1.3835	3.6343	226.6	4.0782	0.0246	424.1
3	29889.	174.90	1.3836	3.6345	227.5	4.0575	0.0244	426.6
4	29748.	176.47	1.3837	3.6336	227.7	4.0494	0.0245	429.8
5	64519.	221.12	1.3980	3.7108	489.1	3.5352	0.0159	1981.0
6	65035.	222.03	1.3981	3.7110	492.9	3.5366	0.0158	1988.9
7	65176.	222.20	1.3981	3.7110	493.9	3.5364	0.0158	1988.2
8	65141.	222.17	1.3981	3.7110	493.7	3.5359	0.0158	1988.9
9	65504.	222.33	1.3981	3.7112	496.4	3.5324	0.0157	1995.1
10	65657.	223.36	1.3981	3.7107	497.4	3.5375	0.0158	1995.8
11	65833.	223.78	1.3982	3.7107	498.6	3.5491	0.0158	2002.7
12	66032.	223.97	1.3982	3.7106	500.0	3.5502	0.0158	2002.7
13	66016.	223.83	1.3982	3.7105	499.9	3.5373	0.0157	2004.6
14	66332.	224.72	1.3983	3.7086	499.4	3.5468	0.0158	2004.2
15	66163.	225.15	1.3983	3.7070	499.9	3.5500	0.0158	2003.7

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MK41
LIQUID OXYGEN TURBOPUMP ASSEMBLY

PROCESSING DATE 06-01-78
TEST DURATION, SEC 43.00

RUN NUMBER 6
TEST DATE 05-31-78

S E A L A N D B E A R I N G D A T A

TIME SLICE NO	PRIM LOX SEAL DR PR (PSIA)	LOX DR SEAL DR PR (PSIA)	I/S PURGE PR (PSIA)	I/S PURGE TEMP (DEG R)	PRIM HG SEAL ORIF U/S PR (PSIA)	PRIM HG SEAL ORIF U/S TEMP (DEG R)	PRIM HG SEAL DR SEAL DR PR (PSIA)	PRIM HG SEAL DR SEAL DR PR (PSIA)	SEC HG SEAL DR SEAL DR PR (PSIA)	SEC HG SEAL DR SEAL DR PR (PSIA)	SEC HG SEAL DR SEAL DR PR (PSIA)	SEC HG SEAL DR SEAL DR PR (PSIA)	SEC HG SEAL DR SEAL DR PR (PSIA)
1	15.55	152.24	214.46	551.37	14.89	272.11	0.0310	16.5	462.89	13.95	441.94		
2	15.45	160.22	214.33	551.39	14.81	273.94	0.0299	16.5	462.12	13.94	435.19		
3	15.47	164.75	214.01	551.69	14.77	274.19	0.0293	16.5	461.41	13.94	430.34		
4	15.47	174.99	213.80	551.72	14.73	274.60	0.0286	16.5	461.71	13.93	426.79		
5	17.36	171.69	215.14	551.72	25.82	353.40	0.0776	21.1	475.95	14.17	426.56		
6	17.37	174.83	215.21	551.72	28.32	367.08	0.0836	21.5	479.41	14.19	427.60		
7	17.45	175.14	215.28	551.72	28.82	368.96	0.0848	21.4	480.58	14.17	427.85		
8	17.44	176.08	215.35	551.72	29.57	374.69	0.0864	21.3	481.64	14.17	428.47		
9	17.44	178.63	215.56	551.72	31.37	386.43	0.0875	21.8	482.93	14.20	429.45		
10	17.58	180.35	215.56	551.72	34.20	398.42	0.0971	23.6	485.03	14.30	431.06		
11	17.55	183.48	215.91	551.63	36.46	418.67	0.1012	24.2	488.64	14.33	434.50		
12	17.57	184.72	216.12	551.72	36.87	425.76	0.1115	24.2	493.15	14.36	436.22		
13	17.56	185.19	216.19	551.72	37.14	428.45	0.1019	24.4	490.73	14.31	437.07		
14	17.56	192.61	216.95	551.72	39.47	448.37	0.1061	24.5	495.22	14.33	445.55		
15	17.58	198.61	217.22	551.72	40.18	456.35	0.1071	24.4	497.52	14.29	451.51		

MK48
LIQUID OXYGEN TURBOPUMP ASSEMBLY

PROCESSING DATE 06-01-78
TEST DURATION, SEC 43.03

RUN NUMBER 6
TEST DATE 05-31-78

SEAL AND BEARING DATA (CONTINUED)

TIME SLICE NO	SEAL DR FLOW (LB/SEC)	RR BRG COOLANT SUPPL. PR (PSIA)	RR BRG COOLANT SUPPL. TEMP (DEG R)	RR BRG DRAIN PR (PSIA)	RR BRG DRAIN TEMP (DEG R)	RR BRG COOL-OR IF U/S PR (PSIA)	RR BRG COOL-OR IF U/S TEMP (DEG R)	RR BRG COOLANT FLOW (LB/SEC)
1	0.0	3519.8	127.69	205.7	92.34	13.8	460.00	0.0
2	0.0	3520.3	127.81	202.7	93.20	13.8	460.00	0.0
3	0.0	3519.6	127.69	201.0	88.89	13.8	460.00	0.0
4	0.0	3519.9	127.75	200.8	88.11	13.8	460.00	0.0
5	0.0	3533.1	127.81	189.3	91.51	13.8	460.00	0.0
6	0.0	3532.4	127.87	190.6	92.43	13.8	460.00	0.0
7	0.0	3533.8	127.87	190.8	92.63	13.8	460.00	0.0
8	0.0	3533.1	128.12	190.4	92.67	13.8	460.00	0.0
9	0.0	3532.4	127.62	191.0	92.75	13.8	460.00	0.0
10	0.0	3534.5	127.62	190.4	92.84	13.8	460.00	0.0
11	0.0	3533.8	127.87	191.0	93.22	13.8	460.00	0.0
12	0.0	3533.8	127.87	191.3	93.31	13.8	460.00	0.0
13	0.0	3533.5	127.75	191.1	93.12	13.8	460.00	0.0
14	0.0	3535.1	127.81	191.5	93.32	13.8	460.00	0.0
15	0.0	3536.0	127.81	192.0	94.29	13.8	460.00	0.0

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MK46
LIQUID OXYGEN TURBOPUMP ASSEMBLY

RUN NUMBER 6
TEST DATE 05-31-78
PROCESSING DATE 06-01-78
TEST DURATION, SEC 43.00

SEAL AND BEARING DATA (CONTINUED)

TIME SLICE NO	LOX SEAL DRAIN ORIF U/S PRESS PSIA	LOX SEAL DRAIN ORIF U/S TEMP DEG R	LOX SEAL DRAIN FLOWRATE (LB/SEC)
1	14.1	520.65	0.093
2	14.0	521.03	0.087
3	14.0	521.33	0.087
4	14.0	521.55	0.086
5	14.4	521.51	0.137
6	14.4	521.46	0.139
7	14.4	521.37	0.137
8	14.4	521.37	0.140
9	14.4	521.37	0.13
10	14.4	521.28	0.140
11	14.4	521.20	0.140
12	14.4	521.20	0.140
13	14.4	521.20	0.143
14	14.4	521.05	0.139
15	14.4	521.04	0.139

MK4.
LIQUID OXYGEN TURBOPUMP ASSEMBLY

PROCESSING DATE 06-01-78
TEST DURATION, SEC 43.00

RUN NUMBER 6
TEST DATE 05-31-78

P U M P P R E S S U R E S

TIME SLICE NO	SUPPLY TANK		PUMP INLET		IMPELLER DISCH		DIFFUSER DISCH		PUMP DISCH		BAL PIST CAV		BAL PIST SUMP		BAL PIST RET'RN	
	PR	(PSIA)	PR	(PSIA)	PR	(PSIA)	PR	(PSIA)	PR	(PSIA)	PR #1	(PSIA)	PR	(PSIA)	PR	(PSIA)
1	267.5		248.6		513.7		830.8		1042.3		775.9		368.3		326.4	
2	267.7		247.9		551.0		783.9		994.3		746.6		362.2		318.7	
3	267.5		247.4		595.7		788.9		999.9		748.5		363.3		319.5	
4	267.3		247.1		593.8		785.3		1001.8		747.3		365.5		320.8	
5	264.7		176.5		2067.5		2971.9		4067.9		2293.6		1206.4		1136.9	
6	264.5		174.3		2097.9		2998.2		4124.6		2305.5		1222.7		1160.9	
7	264.3		175.2		2108.4		3012.1		4142.2		2309.0		1232.6		1170.7	
8	264.2		175.6		2119.0		3034.2		4159.0		2304.1		1252.4		1172.1	
9	264.0		176.0		2113.4		3061.3		4201.2		2324.5		1251.7		1188.2	
10	263.8		169.7		2123.2		3059.9		4190.7		2311.9		1263.6		1195.9	
11	263.8		169.2		2119.7		3047.4		4194.9		2321.7		1268.6		1208.4	
12	263.7		165.2		2125.6		3052.3		4207.6		2319.6		1275.2		1211.9	
13	263.7		169.3		2131.5		3052.5		4206.6		2324.5		1260.0		1212.6	
14	263.9		163.6		2109.3		2999.3		4099.8		2225.4		1265.9		1203.8	
15	264.0		163.5		2115.4		2996.6		4113.1		2216.3					

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MK4L
LIQUID OXYGEN TURBOPUMP ASSEMBLY

PROCESSING DATE 06-01-76
TEST DURATION, SEC 43.00

RUN NUMBER 6
TEST DATE 05-31-78

PUMP PRESSURES AND TEMPERATURES

TIME SLICE NO	SPEED RPM	INDUCER		PUMP BRG	
		DISCH PRESS PSIA	D/S PRESS PSIA	D/S PRESS PSIA	PUMP BRG PRESS PSIA
1	29909.	314.9		310.6	
2	25568.	300.7		305.3	
3	25689.	300.6		309.0	
4	29748.	575.8		1144.9	
5	64519.	578.2		1164.4	
6	65035.	578.2		1165.8	
7	65176.	586.9		1176.4	
8	65141.	592.3		1189.7	
9	65504.	586.9		1188.3	
10	65657.	577.4		1191.9	
11	65833.	573.6		1191.1	
12	66032.	574.5		1195.8	
13	66016.	527.2		1183.2	
14	66032.	527.3		1210.0	
15	66163.				

MK48
LIQUID OXYGEN TURBOPUMP ASSEMBLY

RUN NUMBER 6
TEST DATE 05-31-78
PROCESSING DATE 06-01-78
TEST DURATION, SEC 43.00

PUMP PRESSURES AND TEMPERATURES (CONTINUED)

TIME SLICE NO	BAL PIST		BAL PIST		BAL PIST	
	OVBD PRESS PSIA	TEMP DEG R	OVBD DELTA P PSI	OVBD DELTA P PSI	FLOW LB/SEC	
1	294.4	195.3	70.19		1.712	
2	288.8	194.9	68.32		1.690	
3	290.2	195.1	68.50		1.692	
4	290.8	195.4	68.49		1.691	
5	998.2	260.7	216.79		2.663	
6	1019.0	261.5	221.56		2.683	
7	1030.9	261.8	221.46		2.688	
8	1032.6	262.4	218.77		2.666	
9	1046.9	263.2	219.56		2.666	
10	1055.8	263.6	220.21		2.668	
11	1068.1	264.4	218.97		2.655	
12	1073.6	264.9	219.77		2.657	
13	1073.3	265.0	218.46		2.648	
14	1066.9	265.6	213.11		2.608	
15	1074.2	266.0	213.14		2.606	

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MK48
LIQUID OXYGEN TURBOPUMP ASSEMBLY

RUN NUMBER 6
 TEST DATE 05-31-78
 PROCESSING DATE 06-01-78
 TEST DURATION, SEC 43.00

P U M P P R E S S U R E S A N D T E M P E R A T U R E S

TIME SLICE NO	PUMP INLET TEMP (DEG R)	PUMP DISCH TEMP (DEG R)	BAL PIST RETURN TEMP (DEG R)	PUMP DISCH U/S TEMP (DEG R)		PUMP DISCH U/S PR (PSIA)		PUMP DISCH DELTA PR (PSI)		HP (TEMP)
				VENT	U/S	VENT	U/S	VENT	U/S	
1	171.86	176.04	194.52	176.33	176.33	1034.3	1034.3	8.59	8.59	69.70
2	171.75	175.85	194.73	176.22	176.22	985.6	985.6	9.74	9.74	71.00
3	171.74	175.85	194.61	176.21	176.21	991.0	991.0	9.92	9.92	71.94
4	171.74	175.88	195.12	176.29	176.29	992.5	992.5	9.96	9.96	72.41
5	171.80	191.16	249.75	190.15	190.15	4019.9	4019.9	25.84	25.84	569.81
6	171.79	191.26	253.81	190.34	190.34	4072.9	4072.9	27.41	27.41	591.70
7	171.81	191.38	254.27	193.37	193.37	4091.7	4091.7	28.61	28.61	606.31
8	171.77	191.45	255.69	190.40	190.40	4109.1	4109.1	27.69	27.69	599.77
9	171.79	191.61	258.64	190.70	190.70	4152.3	4152.3	28.20	28.20	610.13
10	171.77	191.71	261.24	193.83	193.83	4139.1	4139.1	29.92	29.92	628.41
11	171.77	191.84	264.95	191.16	191.16	4140.4	4140.4	31.81	31.81	648.50
12	171.74	191.87	266.16	191.19	191.19	4152.3	4152.3	32.19	32.19	654.23
13	171.75	191.92	266.54	191.25	191.25	4149.7	4149.7	32.48	32.48	657.28
14	171.74	191.86	269.66	191.43	191.43	4038.2	4038.2	37.55	37.55	692.42
15	171.74	192.00	270.90	171.61	171.61	4050.2	4050.2	39.19	39.19	709.75

MK42
LIQUID OXYGEN TURBOPUMP ASSEMBLY

PROCESSING DATE 06-01-78
TEST DURATION, SEC 43.00

RUN NUMBER 6
TEST DATE 05-31-78

C A L C U L A T E D P U M P P A R A M E T E R S

TIME SLICE NO	TEST SPEED (RPM)	PUMP FLOW GPM	PUMP FLOW (LB/SEC)	HEAD FT	PRESS RISE (PSI)	FLUID HP	TURE HP (HP)	EFF (%)	ISEN EFF (%)	SPECIFIC SPEED
1	29909.	97.89	15.2342	1654.1	793.8	45.8	93.81	48.86	65.74	1141.
2	29568.	104.26	16.2179	1558.8	746.4	46.0	91.92	50.02	64.74	1217.
3	29689.	105.21	16.3679	1571.8	752.5	46.8	92.51	50.58	65.02	1220.
4	29748.	105.43	16.3996	1576.6	754.8	47.0	93.37	50.37	64.92	1221.
5	64519.	169.16	26.5827	8048.2	3891.4	389.0	645.12	60.32	68.27	988.
6	65035.	174.21	27.3825	8173.3	3950.3	406.8	651.24	62.48	68.75	999.
7	65176.	177.97	27.9777	8207.1	3966.9	417.5	651.51	64.10	68.86	1008.
8	65141.	177.7	27.5265	8238.5	3983.5	412.3	651.37	63.37	68.75	997.
9	65504.	176.08	27.7787	8324.7	4025.3	420.5	653.94	64.32	68.91	999.
10	65657.	182.03	28.6073	8321.1	4021.0	432.8	658.73	65.73	68.87	1017.
11	65833.	187.76	29.4882	8335.8	4025.7	446.9	664.58	67.27	68.92	1034.
12	66032.	189.87	29.6653	8361.7	4038.3	451.0	665.41	67.80	68.94	1038.
13	66016.	189.74	29.7961	8360.8	4037.3	452.9	662.79	68.36	68.91	1040.
14	66032.	204.19	32.0087	8168.8	3936.1	475.4	667.38	71.26	68.66	1098.
15	66163.	208.63	32.6970	8200.4	3949.6	487.5	668.66	72.94	68.69	1109.

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MK46
LIQUID OXYGEN TURBOPUMP ASSEMBLY

PROCESSING DATE 06-01-78
TEST DURATION, SEC 43.00

RUN NUMBER 6
TEST DATE 05-31-78

C A L C U L A T E D P U M P P A R A M E T E R S

TIME SLICE NO	TEST SPEED (RPM)	NPSH (FT)	SUCTION SPECIFIC SPEED	INLET FLOW COEFF	HEAD COEFF	PUMP DELTA T (DEG R)	(O/N) OVER (O/N)DES
1	29909.	463.01	2564.70	0.08318	0.48055	4.18	0.9875
2	29568.	461.57	3031.71	0.08962	0.46340	4.10	1.0639
3	29686.	460.69	3062.48	0.09007	0.46342	4.11	1.0692
4	29748.	459.91	3075.60	0.09078	0.46300	4.14	1.0694
5	64515.	317.03	11169.17	0.06664	0.50246	19.36	0.7911
6	65035.	312.73	11542.78	0.06908	0.50203	19.48	0.8082
7	65176.	314.86	11632.48	0.06940	0.50212	19.57	0.8239
8	65141.	315.41	11515.90	0.06831	0.50458	19.67	0.8109
9	65504.	216.32	11608.46	0.06856	0.50422	19.82	0.8138
10	65657.	303.66	12177.64	0.07347	0.50166	19.93	0.8365
11	65833.	302.81	12426.90	0.07249	0.49987	20.07	0.8605
12	66032.	302.94	12497.24	0.07270	0.49840	20.13	0.8630
13	66016.	303.20	12514.95	0.07305	0.49558	20.17	0.8672
14	66032.	292.27	13348.57	0.07860	0.48690	20.12	0.9330
15	66163.	292.23	13520.99	0.08015	0.48635	20.25	0.9514

MK44
LIQUID OXYGEN TURBOPUMP ASSEMBLY

RUN NUMBER 6
TEST DATE 05-31-78
PROCESSING DATE 06-01-78
TEST DURATION, SEC 43.00

SCALED TO TARGET SPEED = 70000. RPM

TIME SLICE NO	FLOW (GPM)	FLOW (#/SEC)	HEAD (FT)	PRESS RISE (PSI)	HORSE POWER (BHP)	NPSH (FT)
1	229.19	35.65405	9060.37	4347.79	893.46	2536.10
2	246.82	38.35525	8736.96	4183.32	942.11	2587.04
3	248.06	38.55141	8737.32	4183.10	942.92	2560.96
4	248.99	38.58981	8729.54	4179.23	943.50	2546.56
5	183.53	28.84074	9473.52	4580.57	727.69	373.18
6	187.51	29.47278	9465.32	4576.42	737.81	362.29
7	191.14	30.34841	9466.96	4575.87	751.15	363.19
8	188.13	29.57978	9513.40	4599.91	744.24	364.22
9	188.81	29.58523	9506.61	4596.74	744.57	361.23
10	194.07	30.49672	9458.41	4570.55	761.56	345.16
11	199.65	31.35491	9424.54	4551.56	779.62	342.36
12	200.22	31.44774	9396.90	4538.29	779.40	340.45
13	201.19	31.59413	9400.28	4539.28	783.60	343.90
14	216.46	33.92228	9180.12	4423.43	824.93	328.45
15	220.73	34.59333	9179.17	4421.04	840.54	327.11

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MK4
LIQUID OXYGEN TURBOPUMP ASSEMBLY

PROCESSING DATE 06-01-78
TEST DURATION, SEC 43.00

RUN NUMBER 6
TEST DATE 05-31-78

SCALED TO DESIGN SPEED = 70,000. RPM

TIME SLICE NO	FLOW (G)	FLOW (M/SEC)	HEAD (FT)	PRFSS RISE (PSI)	HORSE POWER (BHP)	MPSH (FT)
1	229.39	35.65405	9060.37	4347.79	893.46	2536.10
2	246.82	38.39525	8736.96	4183.32	942.11	2587.04
3	248.06	38.55141	8737.32	4183.10	942.92	2560.96
4	248.39	38.58981	8729.54	4179.23	943.53	2546.56
5	183.53	28.84074	9473.52	4580.57	727.69	373.18
6	187.51	29.47278	9465.32	4576.42	737.81	362.29
7	191.14	30.36841	9466.96	4575.87	751.15	363.19
8	188.13	29.57978	9513.40	4599.91	744.24	364.22
9	188.81	29.65523	9506.61	4596.74	744.57	361.23
10	194.07	30.43972	9458.41	4573.55	761.56	345.16
11	199.65	31.35491	9424.54	4551.56	779.62	342.36
12	200.22	31.44774	9396.90	4538.29	779.40	340.45
13	201.19	31.59413	9400.28	4539.28	783.60	343.90
14	216.46	33.93228	9180.12	4423.43	824.90	328.45
15	220.73	34.59333	9179.17	4421.04	840.54	327.11

REFERENCES

1. Interim Report, "Small, High-Pressure Liquid Oxygen Turbopump," by A. Csomor and R. Sutton; NASA CR 135211 Rocketdyne No. R76-178.
2. Young, W. E. and Dul, H. F., "Investigation of Pressure Prediction Methods for Low Flow Radial Impellers," Draft, PWA FR-1716, 13 January 1966.